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THE *WARRINGTON'S* COLLISION.

BY LIEUTENANT COMMANDER W. M. HUNT, U. S. NAVY, COMMANDING.

The U. S. destroyer *Warrington* was struck by an unidentified schooner, in a winter gale off Hatteras, at 12:40 A. M., December 28, 1911, and 30 feet of her stern was cut off and sank.

The *Warrington,* in formation with seven other destroyers, was making passage from Charleston, S. C., to New York, to take part in maneuvers with the battleship fleet, on the way South to the Winter rendezvous.

Shortly after leaving Charleston the wind freshened to a south-west gale, and soon the destroyers were laboring in a heavy following sea, which made the vessels yaw constantly and roll uncomfortably. During the afternoon and evening, the wind hauled to the westward; and when Diamond Shoal lightship, off Cape Hatteras, was reached, about 11:30 P. M., the wind was blowing a moderate gale from the west, with a

correspondingly heavy sea; and a change of course to the northward did not help matters much.

At about 12:40 A. M., a red light was suddenly observed close aboard on the starboard bow, and, although every effort was made to avoid a collision, the approaching vessel, which then appeared to be a schooner, struck the *Warrington* on the starboard quarter, the head booms carrying away the main mast and wireless aerial, and the schooner's bow cutting into and shearing off the entire stern; thirty feet was completely detached and went to the bottom. The schooner passed on as quickly as she had come, and disappeared in the darkness of the night.

It is remarkable that, although the shock at both ends of the ship was severe, the men on watch in the fireroom did not feel anything, and knew nothing about the collision until they were told of it.

In the compartment which the schooner struck and opened to the sea six men were asleep in their bunks. Two of these men made their escape out over the jagged side; three opened the bulkhead door into the next compartment forward, and escaped to the deck together with thirteen other men who had been asleep there; the sixth man was injured and rendered half unconscious by the collision, and while in his semi-conscious state, managed to hook his arm around a stanchion. Both of these compartments immediately became flooded to a depth of about three feet; the water, surging violently with the roll and pitch of the ship, quickly broke up the wooden lockers and mess tables, and detached the bunk frames from their fastenings; and the wreckage was thrown from side to side with the rush of the water.

Immediately after the collision the officer of the deck rushed aft to ascertain the extent of the damage. Finding water in the compartment forward of the one cut open, he went into that compartment, followed by one of the crew, in an attempt to close the bulkhead door through which three of the crew had escaped, in their bewildered and half-asleep condition, and had left open. Forcing their way through the rush of water

and wreckage to the door, they heard moans, and, searching, found the half unconscious man, who was being covered with water at every roll of the ship. With considerable difficulty they got him to the ladder, and, with further assistance, on deck. The officer then went back to the door, but the dogs had become jammed and bent as the rush of the water swung the door from side to side, and the door could not be closed.

The water, filled with wreckage, was now pounding against the collision bulkhead at the after end of the engine room, some forty feet forward of where the stern had been cut off, and the impact of the water and wreckage on the bulkhead could plainly be felt on the engine-room side. The little lumber on board had been carried away by the collision, but small odd pieces were secured (even the cook's chopping block was cut up), and the weakest points in the bulkhead shored up, while the circulating pumps were held ready to put on the bilge suction.

A muster of the crew showed all accounted for. Life preservers had been served out, and nearly every man wore one. Two men injured in the collision—the one taken from the after compartment, and another, who cut his foot badly climbing out over the jagged stern—were taken into the officers' quarters, not only to make them as comfortable as possible, but to remove their depressing influence from the rest of the crew. As soon as it was assured that the ship would not sink immediately, the starboard anchor was let go in 18 fathoms of water, a bearing of Cape Hatteras light taken, and the position plotted. This fixed the ship about 17 miles east north-east of Hatteras. The ship rode beam to the sea and, the hard bottom giving no hold, the anchor dragged continually.

Although the night was clear, it was very dark, and that fact, combined with the effort of holding on to the swaying bridge rails, apparently prevented the other vessels from observing the plight of the *Warrington*. Signals of distress were made by whistle and electric night signals; meanwhile the lights of the other ships were fast disappearing to the

northward, and for a few minutes it looked as if the ship would have to work out her own salvation; but the welcome sight of a destroyer's running lights was shortly discovered, returning to the *Warrington's* position.

The port chain was unbent from the anchor, and the six-inch tow line got up and bent to the chain, and heaving and running lines prepared to pass to the destroyer. The destroyer passed close along the *Warrington's* side in a gallant attempt to take the line; but the violent motion of both, and the heavy wind, made it impossible to get the line aboard. It was an exhibition of most skilful handling of the other destroyer, as she passed close alongside, at slow speed, with screws partly out of the water at times, where an error of judgment in the darkness would have been serious for both vessels.

The impossibility of passing the towline at night being apparent, the wireless of the other destroyer was called into play, and just before dawn two more destroyers returned.

It has often been said by watchers in the night that the longest hours are those just before dawn. The feeling on board the *Warrington* was no exception to this. The crew were kept occupied with various duties; galley fires started, and coffee served out. Frequent visits about the ship showed the crew hopeful; but there was a tense undercurrent—of not knowing but that an indefinite something might happen.

Daybreak showed no change in the weather, and the wind seemed, if anything, stronger. The anchor chain was riding more up and down than in the night, and frequent soundings showed that the ship was drifting constantly to seaward. The other destroyers steamed slowly around, rolling and tossing apparently much more than the *Warrington*. The decks of their bridges would appear for a moment, with the officers and men holding on to the rail, and the next instant the turn of the bilge would be presented to view, with occasionally a propeller churning the surface of the water. Now and then a destroyer would steam by to windward and put out an oil slick, but by the time it reached the *Warrington* it was dissipated over so much of the sea that the effect was nullified.

After daylight it was found that the water was getting higher in the compartment abaft the engine room. Several times volunteers with bowlines were sent in along the top of the stanchions over the surging water and wreckage, in attempts to close the bulkhead door, swinging wildly from side to side with the water. After desperate work one of the bent dogs was frequently cleared and the door closed, only to be forced open on the next roll. Pieces of wood were fished out of the wreckage to remove some of the battering on the collision bulkhead, and the larger pieces were sent into the engine room to add to the shoring.

A cork anchor buoy was bent to a coil of signal halliard stuff and veered over the lee bow, with a length of three-inch line, and finally the entire length of six-inch hawser; and attempts were made by the other destroyers to pick up the buoy; but they only succeeded in carrying it away. An attempt was made also to trail a buoy across the bow of the *Warrington*, but the anchor was then beginning to trip, and the ship drifted so fast that the buoy remained well to windward. During this attempt the stern of the other craft passed so close that heaving lines were thrown by both vessels, only to be frustrated by the heavy wind, and the violent motion of the two vessels. Seas swept over the stern of the other destroyer and all but carried several of her crew overboard.

At this time the ship was close to the 100-fathom curve, and still drifting to seaward. The anchor would occasionally strike a deep spot and lose contact with the bottom; and the ship showed a tendency to bring her stern up to the sea, which would have placed the collision bulkhead in serious danger of being carried away.

The bridge screen was taken down, and an awning rigged between the after stacks as a steadying sail, but the wind and sea on the high bow were gradually swinging the stern around. At this juncture a revenue cutter, which proved to be the *Onondaga*, was sighted bearing down from the north north-west. In a short time she ranged alongside and offered assistance, and was requested to take the *Warrington* in tow.

It being considered inadvisable, and in fact exceedingly dangerous, for the crew to remain on board while towing, the revenue cutter lowered two boats and transferred 70 of the crew to the cutter, twelve of the crew and three officers remaining on board. Oil was freely used by both vessels, without which it is believed the transfer could not have been made, or, at least it would have been extremely dangerous. Long sea painters were led out on the lee side of the *Warrington*, and oil poured over the lee bow. Fuel oil in buckets, lubricating oil in cans, and even kerosene oil from tanks in the bow were used. A bag filled with rope yarns was saturated with oil and placed over the side, but it did not cover enough surface, and it was found necessary to spill a pint or so of oil from cans or buckets every half minute. (As a considerable quantity of this oil was washed or blown against the side and on deck, only a small quantity became effective, and it is believed that a very small quantity, judiciously handled and placed, would have given the desired effect.) There was not much opportunity to compare the effect of fuel, with lubricating oil. Both were effective. Kerosene was too light to be of much use.

The staunch "Monomoy" life boats of the cutter, patterned after the boats used by the life-saving service—unsinkable and with plenty of beam; manned by a coxswain and six men —were capable of carrying ten or twelve passengers. When near enough, several of the *Warrington's* crew hurdled into the boats, the crew receiving them as best they could, and the boats would swing out and back again to receive more of the crew. After several trips of the boats to transfer the crew, one came under the lee bow and took a light line, by means of which the tow line was hauled on board the cutter. The starboard anchor was then hove up, and the tow straightened out for port. Fortunately, the wind and sea were such that with both on the bow a course could be laid for the capes of the Chesapeake. The ship rode easily, head to the tow line, due, it was afterward found, to the drag of both propeller shafts, which were hanging down from the stern tubes.

Soon after starting the tow, the weather began to moderate,

and continued to moderate until reaching port. Being, however, over a hundred miles from the capes, and making only 5 to 7 knots an hour, there was ample opportunity for the weather to change, with the possibility of heavy seas endangering the collision bulkhead; and a close watch was kept throughout the night. Early the next forenoon, the sea having moderated somewhat, the bulkhead door was closed and secured, and the water pumped out of the compartment abaft the engine room, thus relieving the pressure on the engine-room bulkhead. The tow was stopped, thirty fathoms of chain was paid out on the tow line, and the tow speeded up to about seven knots.

Upon arrival at Hampton Roads, at about 3:00 P. M., Dec. 30th, the *Onondaga* was relieved by a navy-yard tug, and the *Warrington* taken to the navy yard. With the aid of a diver, the bent shafts were raised, and the ship placed in dry dock. The illustration shows the appearance of the stern as the vessel reached the dock.

Both shafts were badly bent abaft the stern tubes, all struts had been broken off, the port propeller gone, and the blade tips of the starboard propeller were bent and broken. All of the ship abaft frames No. 148 and 149 was entirely gone.

The shape and condition of the fractured metal at frame No. 148, together with the statement of one of the crew who had escaped from the after compartment, leads to the conclusion that the schooner's cutwater first punched a hole through the side about one-third the width of the ship; the two craft then separated, the *Warrington* lying in the hollow of a wave, while the schooner, riding on the crest, bore down on the fractured stern; her bow, combined with the speed and divergence of the two vessels, forming a leverage which sheared off the remaining two-thirds of the hull.

The engines suffered no injury; thrust and steady bearings and line shafting show no distortion. An entire new stern will be manufactured and installed, after which the ship will be ready to take her place with the rest of the destroyer fleet.

REPORT OF SHOP TESTS OF THE STARBOARD PROPELLING UNIT OF THE U. S. S. *HENLEY*.

A COMBINATION OF RECIPROCATING ENGINE AND TURBINE DESIGNED FOR MAXIMUM ECONOMY AT CRUISING SPEED.

The tests were conducted by a board composed of Captain C. W. Dyson, U. S. Navy, and Lieutenant Commander John Halligan, Jr., U. S. Navy, assisted by Messrs. Angelo Conti and Ernest Janson, from the Bureau of Steam Engineering. The board reports as follows:

The unit tested consists of a combination of a reciprocating engine and a turbine, intended to afford a maximum of economy at cruising speeds. The turbine is of the Curtis marine type with 18 stages, 63-inch pitch diameter, and is designed to develop 5,500 S.H.P. at 585 r.p.m., corresponding to a speed of vessel of about 29½ knots. Forward of the turbine, and connected to it by means of a jaw clutch, is a 10½ by 22 by 10-inch vertical, compound reciprocating engine, which at 16 knots is intended to develop 400 I.H.P. at 280 r.p.m., with a steam pressure of 250 pounds gage at the H.P. valve chest and when exhausting into the turbine at about atmospheric pressure. Photographs showing sectional arrangements of the turbine and the cruising engine are shown on Plates I and II. The testing plant was contained in a wooden structure alongside the power house of the shipbuilding company, and contained, in addition to the starboard main turbine and cruising engine of the *Henley*, her starboard main condenser, main circulating pump, main air pump, ahead and astern throttles, and main steam piping. The arrangement of the testing plant and details of the installation are shown on Plate III.

A Froude patent water dynamometer was used to absorb and measure the power generated, and there were installed on

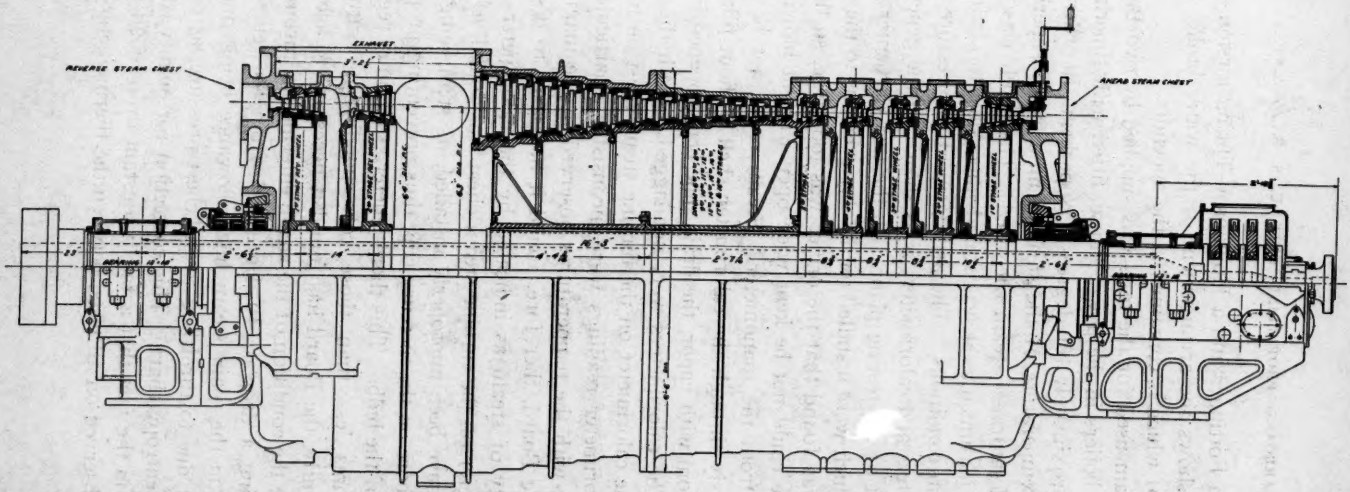


Plate I.—ARRANGEMENT OF TURBINE.

the shaft a Föttinger and a Hopkinson-Thring torsion meter. Plate IV shows the arrangement of the water brake, a description of which is contained in the appendix.

The steam used for the tests was supplied by two Babcock & Wilcox boilers in the power house, fitted with superheaters giving approximately 100 degrees F. superheat. In order to reduce the temperature of the steam a mixture of superheated steam and saturated steam was used. The result was unsatisfactory, inasmuch as it was found impossible to obtain uniform steam conditions. The arrangement of pressure gages, thermometers and calorimeters is as shown on the general arrangement of the testing plant. In the two Carpenter calorimeters, which were installed on the day previous to the official tests, it was found that the orifice was too large, so that the inlet valve could not be kept wide open without blowing the mercury from the manometer. Steam could not be blown continuously through the calorimeter shell without filling the testing room with vapor, the drain pipe being opened into the room. The calorimeters were not lagged. With the exception of the calorimeter on the turbine steam chest on test No. 2, all calorimeter readings were inconsistent, particularly as compared with the temperatures observed in the main steam line. The Board, therefore, found it necesasry to disregard the quality of steam as indicated by the calorimeters. The thermometer on the turbine steam chest was found, after the tests, to have been improperly installed, its stem being larger than the bore on the mercury cup, thus preventing proper immersion of the bulb. This thermometer at all times registered temperatures less than those corresponding to saturation. Consequently, the Board has found it necessary to arbitrarily determine the condition of the steam entering the turbine chest by allowing a drop of 40 degrees F. from the temperature observed in the main steam line, allowance being made for superheat due to throttling to the chest pressure, as obtained from an entropy chart. This correction for quality of steam is based on the fact that a fairly constant drop of 20 degrees F. was observed on all tests, between the thermometers in the

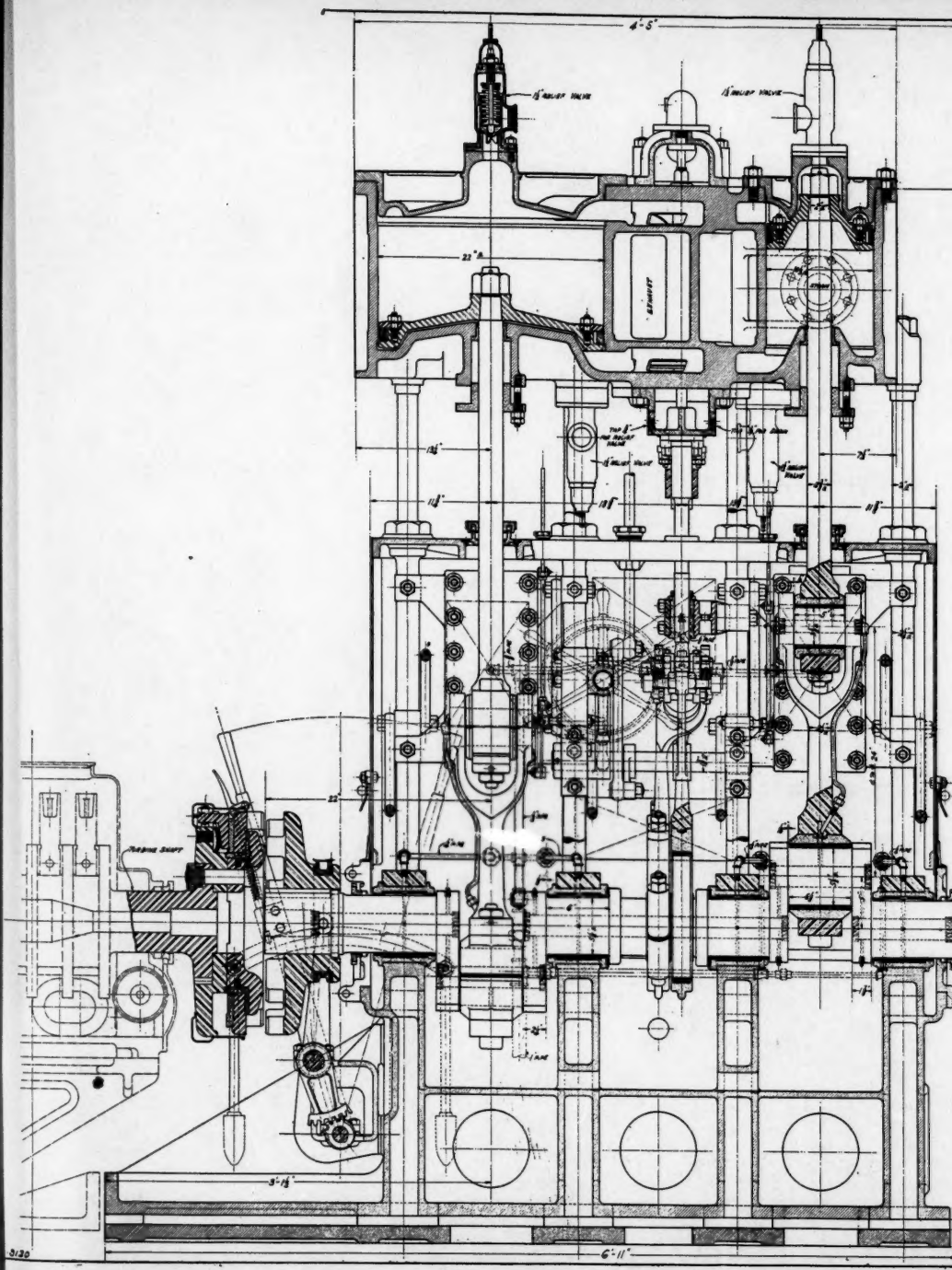
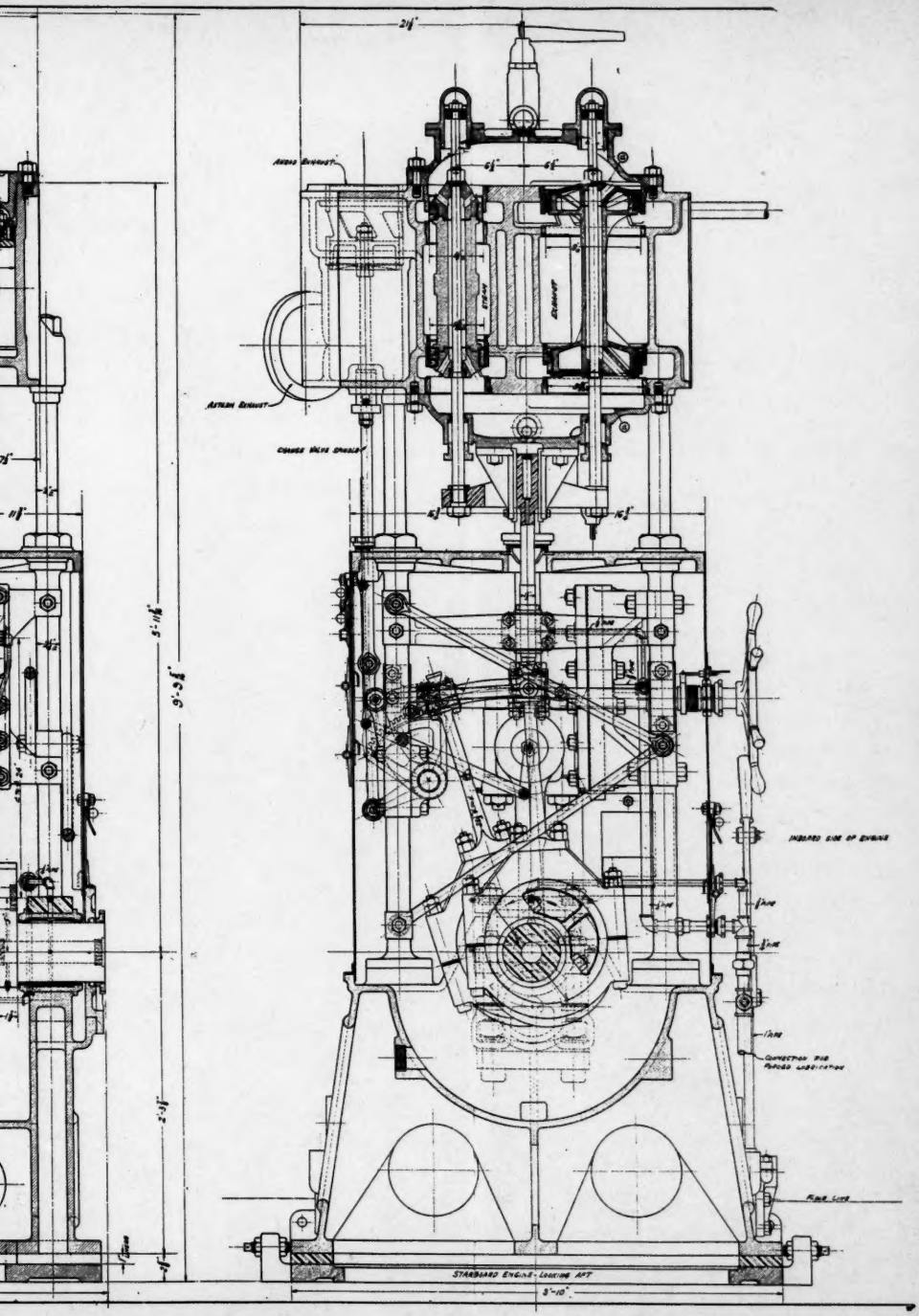


Plate II.—Arrangement of Cru



NT OF CRUISING ENGINE.

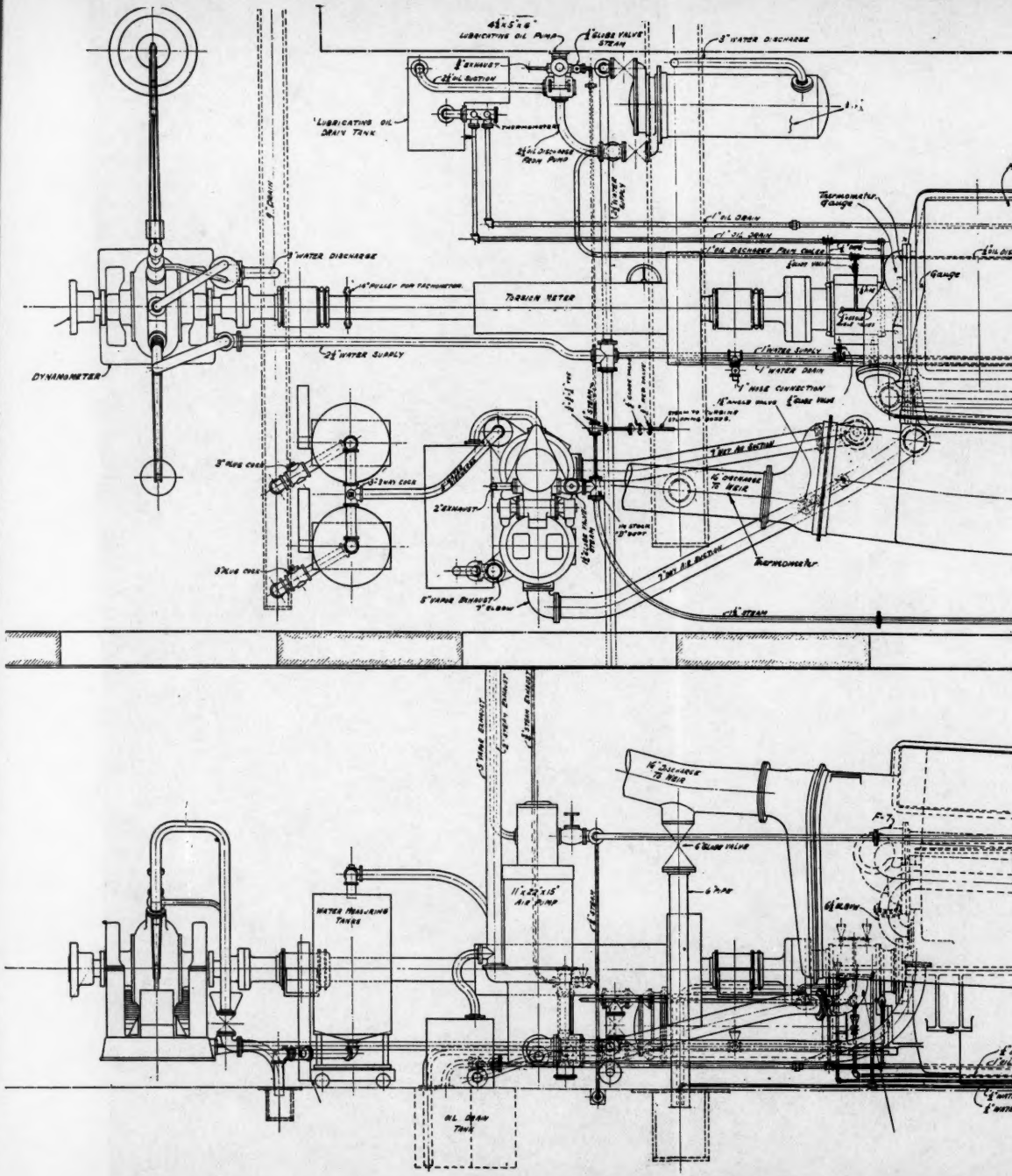
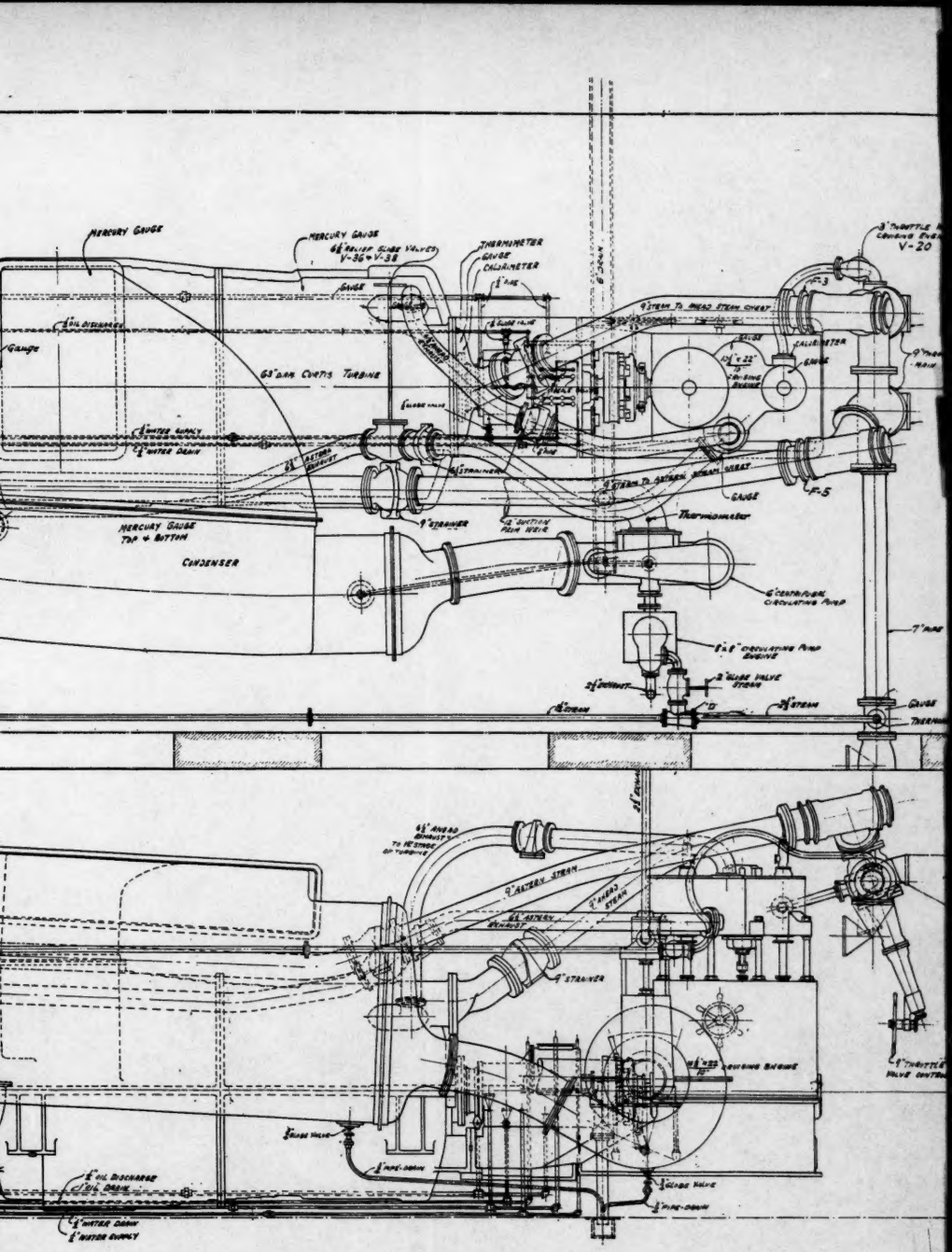


Plate III.—GENERAL ARRANGE



ARRANGEMENT, TEST PLANT.

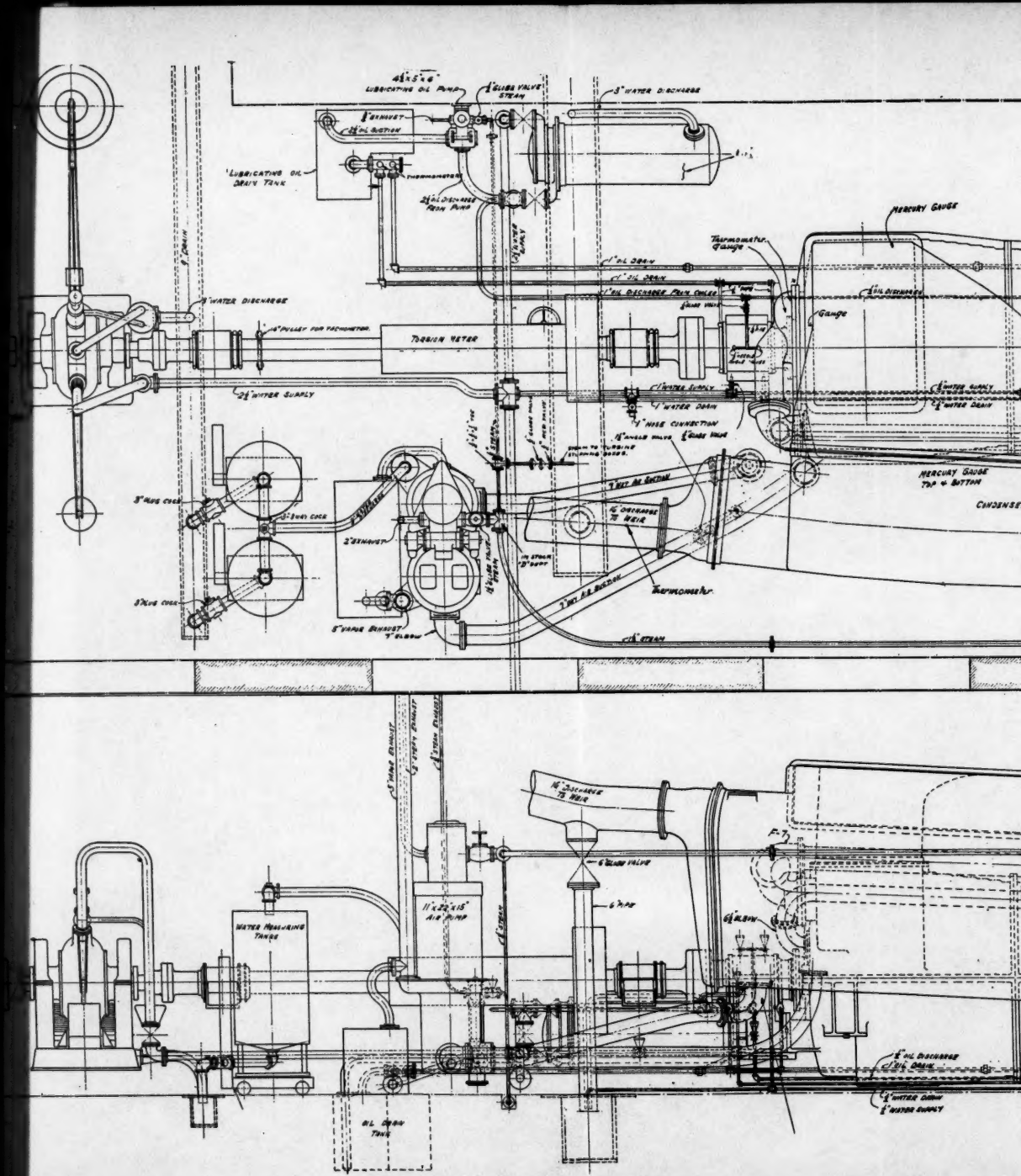
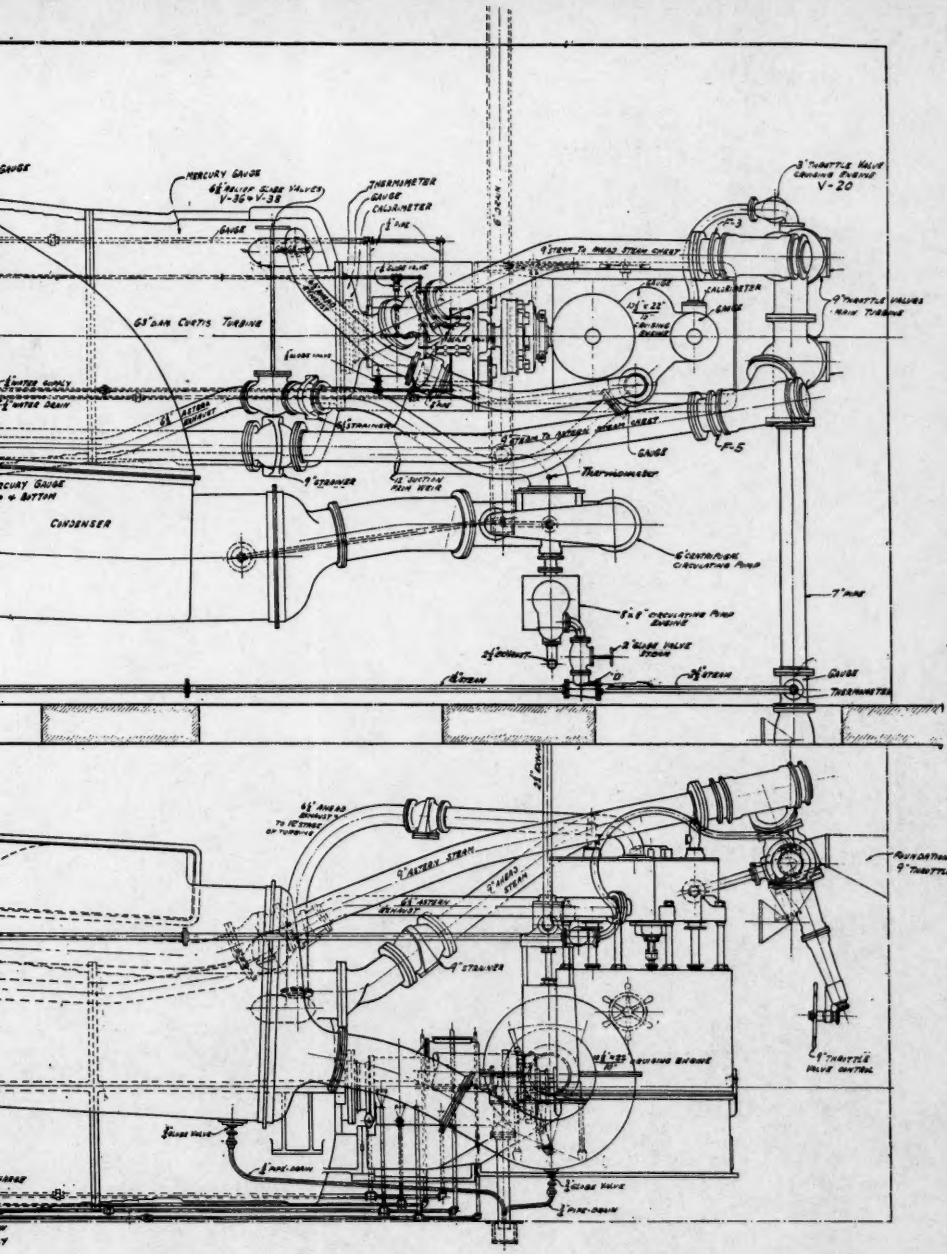


Plate III.—General Arrangement, Test



NT, TEST PLANT.

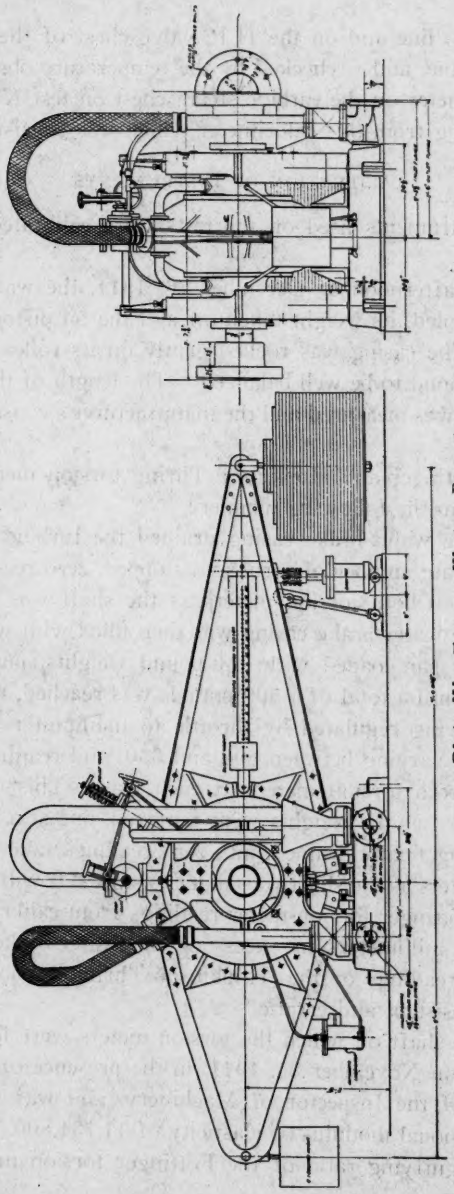


Plate IV.—Froude Water Brake.

main steam line and on the H.P. valve chest of the reciprocating engine, and is checked by the temperature observed on the calorimeter on the turbine steam chest on test No. 2 (the only reading from this calorimeter which was consistent).

CALIBRATION OF INSTRUMENTS.

The instruments used on the tests were calibrated as follows:

On the afternoon of December 26, 1911, the water brake was uncoupled, its weight taken off and the oil piston disconnected. The casing was rocked gently on its roller bearings and was found to be well balanced. The length of the water-brake arm was measured and the manufacturer's constant verified.

The Föttinger and Hopkinson-Thring torsion meters were calibrated in the following manner:

With the water-brake casing drained the turbine was run at a moderate number of turns and stopped, zero readings being taken on both torsion meters as the shaft was about to stop. The water-brake casing was then filled with water and the weight pan loaded with 150-pound weights, put on one at a time until a total of 1,500 pounds was reached, the steam pressure being regulated by throttle to maintain revolutions per minute varying between 300 and 350, and readings being taken on both torsion meters at each load. This operation was repeated as the weights were removed, one at a time, the casing being finally drained, and zero readings taken of both torsion meters as the shaft was about to stop. It will be noted that the Föttinger torsion-meter readings from calibration are about two millimeters in excess of the water-brake record, while the readings of the Hopkinson-Thring torsion meter were inconsistent and erratic.

The line shaft on which the torsion meters were fitted was calibrated on November 25, 1911, in the presence of a representative of the Inspector of Machinery, and was found to have a torsional modulus of elasticity of 11,731,800.

The magnifying ratio of the Föttinger torsion meter was

U.S.S. H

OFFICIAL SHOP T

SUMMARY OF

DATE	TEST NO.	TIME FROM	TIME TO	BAROMETER	MAIN STEAM PRESS. LBS. G.	MAIN STEAM SUPER HEAT °FAH	ENGINE H.P. CHEST PRESS. LBS. G.	ENGINE H.P. CHEST QUAL. OR SUP'HT	ENGINE L.P. REC. LBS G.	ENGINE EXHAUST LBS ABS	TURBINE NO. OF NOZILES OPEN	TURBINE STEAM CHEST PRESS LBS ABS	TURBINE STEAM CHEST QUAL OR SUP'HT	TURBINE 1ST STAGE LBS ABS	TURBINE 5TH STAGE LBS ABS	TURBINE EXHAUST LBS ABS	CONDENSER	R.P.M. BY COUNTER	R.P.M. BY TACH
								Sat. Cal.		AT ENG. AT TURB.			@ Cal.						
DEC. 27TH 1911	1	7:42 PM	9:30 PM	29.5"	244.5	0	202	5.4° 974	42.3	11.15 4.04	—	—	@ Cal.	9.62	1.82	.825	27.82	231	—
" "	2	11:00 "	12:00 "	29.54	244.5	49°	—		—		2 CRUIS	230.9	16° 16°	—	3.10	.953	27.68	235	—
DEC. 28TH 1911	3	2:00 AM	3:00 AM	29.64	242	62.5°	—		—		3 "	220	30° 8°	22	4.2	1.01	27.59	290.2	—
" "	4	4:30 "	5:30 "	29.66	247.2	32.5°	—		—		2 "	146.7	19° 2016	11.6	2.07	.835	27.97	176.2	—
" "	5	7:42 PM	9:00 PM	30.05	247.5	0°	132.5	10° 981	22.15	6.28 6.76	—	—	—	4.93	1.28	.923	28.17	174.3	—
" "	6	9:18 "	10:48 "	30.08	245.3	59°	238.3	46° 986	53.2	16.6 14.77	1 CRUIS	88	—	14.77	2.98	.933	28.18	284	—
" "	7	11:06 "	11:36 "	30.09	245.5	75°	235	65.5 997	46	14.33 12.27	—	—	—	12.3	2.63	.928	28.21	284	—

NOTES: @: SUPERHEAT AS SHOWN BY ENTROPY CHART IN THROTTLING MAIN STEAM TO CHEST PRESSURE, MINUS A DROP OF 40°FAHR.

*: ASSUMED SAME AS TEST #7

Table I.

U.S.S. HENLEY

AL SHOP TEST OF STB'D UNIT

UMMARY OF RESULTS

CONDENSER V.	R.P.M. BY COUNTER	R.P.M. BY TACHOM	CORRECTED T.M. READINGS FOTTINGER	CORRECTED T.M. READINGS HOPKINSON AND THRING	WATER BRAKE WEIGHT	STEAM FLOW LBS/HR	S.H.P. BY T.M. FOTTINGER	S.H.P. BY T.M. HOPKINSON AND THRING	B.H.P.	I.H.P. H.P. CYL.	I.H.P. L.P. CYL.	I.H.P. TOTAL	WATER RATE ENG. LBS/I.H.P.	WATER RATE TURB. LBS/S.H.P.	WATER RATE COMB. LBS/S.H.P.	REMARKS
27.82	231	—	25.4	16.99	1052	6040	385	330	347	126.9	147.3	274.2	22.01	—	17.4	ENG. EXHAUST IN 1ST STAGE SHELL
27.88	235	—	25	15.25	1050	9942	3653	348.6	352.5	—	—	—	—	28.2	—	ENGINE UNCOUPLED
27.59	290.2	—	36.5	—	1500	13854	689	—	622	—	—	—	—	22.27	—	" "
27.97	176.2	—	17	—	640	6392	186.2	—	161	—	—	—	—	39.7	—	" "
28.17	174.3	—	15.5	—	640	3291	165	—	159.4	68.8	81.2	150	21.93	—	20.65	ENG. EXHAUST IN 1ST STAGE SHELL
28.18	284	—	36.5	—	1500	10045	6474	—	609	184.3	196.3	380.6	19.00	—	16.53	ENG. EXHAUST IN 1ST STAGE SHELL LIVE STEAM ADMITTED TO TURBINE
28.21	284	—	29.3	—	1200	7094	517	—	487	190.9	182.6	373.5	18.00	—	14.57	ENG. EXHAUST IN 1ST STAGE SHELL

Table I.

found by calibration to be 47.606, and its constant, .0622. The constant for the Hopkinson-Thring meter, figured from the modulus of elasticity, is .0974.

The spring gages were calibrated on December 15, 1911, in the presence of Chief Machinist Joseph Fitton, U. S. N. On December 27 the water-measuring scales were calibrated.

OFFICIAL TESTS.

All data were checked by members of the Board, assisted by Chief Machinist Joseph Fitton, and by enlisted men in the Navy detailed by the Inspector of Machinery for this purpose.

The first test was intended to represent 13-knot conditions with the reciprocating engine and turbine in combination. This test was begun at 7:42 P. M., December 27, and continued until 9:30 P. M., when one of the link-block bolts of the cruising engine sheared off and the test was discontinued. The engine was uncoupled and tests Nos. 2, 3 and 4 were conducted with the turbine only, under 10, 13 and 16-knot conditions, until 5:30 A. M., December 28. The damage to the cruising engine having been repaired, tests of the combination were resumed on the evening of December 28, beginning at 7:42 P. M. and ending at 12:48 A. M., December 29.

As noted in the appended summary of results, Table I, there were seven official tests in all, one series of three tests with the cruising engine in series with the turbine, and one series of three tests with the engine disconnected, each series being intended to represent 10, 13 and 16 knots, respectively. The r.p.m., and S.H.P. for these speeds being taken from the contractors' estimated curves of performance for the vessel. In the 16-knot condition in the vessel it is contemplated to admit auxiliary exhaust steam into the first-stage shell of the turbine, and, since no such steam was available during the shop trials, it was found necessary in order to obtain the required S.H.P. to admit live steam into the ahead steam chest of the turbine through one cruising nozzle, the pressure being throt-

U. S. S. HENLEY.
SHOP TEST OF STARBOARD UNIT.
COMPARISON OF RESULTS.

Test No.	Duration of Test	Main Steam			Steam at H.P. Cyl. or Turbine Chest		Exhaust Press. at Turbine lbs. abs.	R.P.M.	B.H.P	Total Steam lbs. hr.	Actual W.R : lbs. BHP	Actual: Adiabat Drop BTU's	Actual Thermal Eff. %	Adiabat. Drop to 28.5" of Vacuum BTU's	W.R. Correct. to 28.5" of Vacuum lbs. BHP.	% Improvement of Combination over Turbine			Remarks
		Press. lbs. abs.	Supht. °Fah.	BTU's per lb.	Press. lbs. abs.	Supht. °Fah.										Actual	Corrected to 28.5" of Vac.	Same Vac. & Main Steam Cond.	
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	
4	1 hr.	261.8	32.5°	1225.6	146.7	16°	.835	176.2	161	6392	39.7	327.6	19.57	333.6	39.0	-	-	-	10 knot conditions Turbine only
5	1 hr 18m	262.5	0°	1202.2	147.3	10°	.923	174.3	159.4	3291	20.65	322	38.28	333	19.96	92.2	95.4	98.96	Cruis. eng.& Turb.
2	1 hr.	259.0	49°	1235.8	230.8	16°	.953	235	352.5	9942	28.2	350	25.8	362	27.25	-	-	-	13 knot conditions Turbine only
1	1 hr 48m	259	0°	1202	216.5	5.4°	.825	231	347	6040	17.4	348.5	42.0	354	17.13	62.1	59.1	62.4	Cruis.eng.& Turb.
3	1 hr	256.5	62.5°	1243.4	220	30°	1.01	290.4	622	13854	22.27	347	32.9	362.7	21.30	-	-	-	16 knot conditions Turbine only
6	1 hr 30m	260.1	59°	1241.8	253.5	46°	.933	284	608	10045	16.53	364.1	42.3	375.2	16.04	34.72	32.8	33.0	Cruis.eng.& Turb.
7	30 m.	260.8	75°	1250.7	249.8	65.5°	.928	284	487	7094	14.57	367.8	47.5	378.8	14.05				(Special Test (Cruis.eng.& Turb.

EXPLANATORY NOTES:

Column 14, Thermal Eff.% = $\frac{2545}{\text{W.R.x Adiab.Drop}}$

Column 16, W.R.corrected to a common vacuum of 28.5" (.735# abs.) = Column 12 $\times \frac{\text{Column 13}}{\text{Column 15}}$

Column 17, Based on Column 12

Column 18, Based on Column 16

Column 19, % corrected to common main steam conditions (265# abs.dry) and a common vacuum of 28.5" (.735# abs.) based on W.R. of Column 16 x $\frac{\text{Values of Column 5}}{1202.35}$ (1202.35 = Heat contents of 1 lb. of dry saturated steam at 265# abs.)

Values of Column 19, thus corrected, will be proportional to the actual saving in fuel.

It was deemed undesirable to correct W.R. for steam at chest conditions on account of the considerable throttling.

Table II.

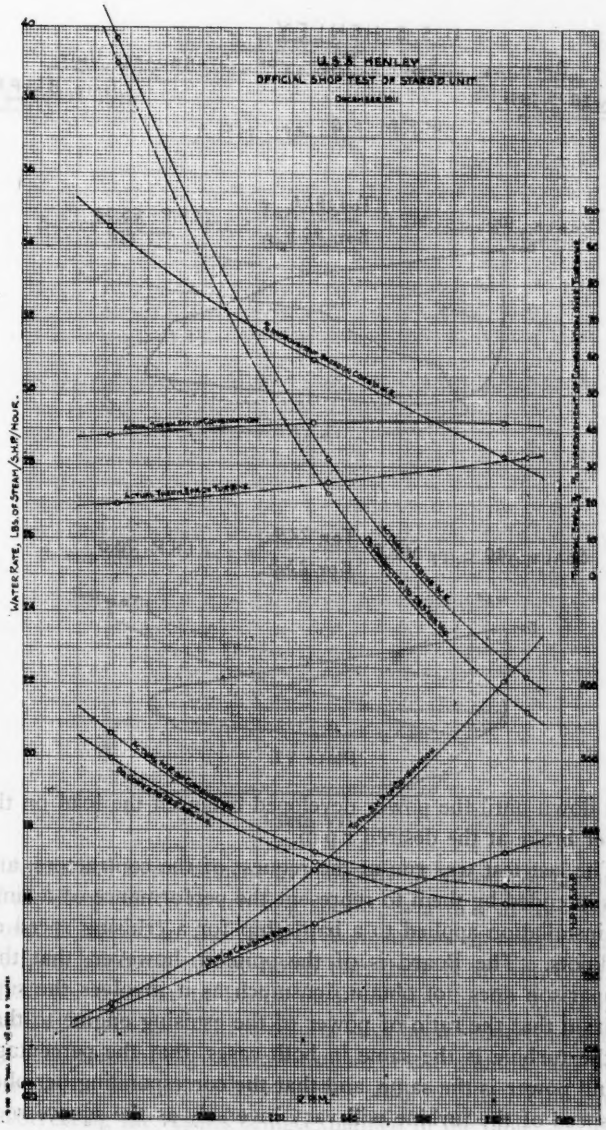

U.S.S. HENLEY
OFFICIAL SHOP TEST OF STARB'D UNIT
WATER RATE, LBS. OF STEAM/S.H.P./HOUR.
R.P.M.

Plate V.

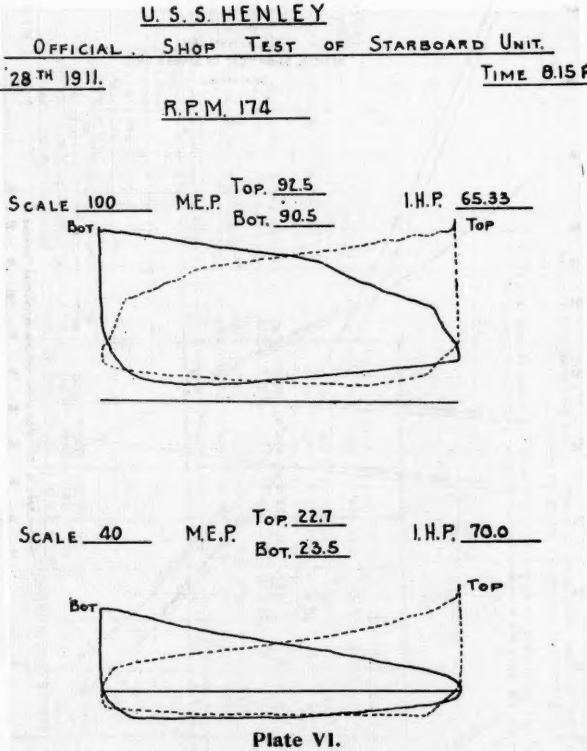


Plate VI.

tled down until the power developed balanced the load on the water brake at the desired r.p.m.

The last test was run at the request of the contractors, and was intended by them to represent the performance of a similar installation applied to a battleship for a cruising speed of 10 knots. The Board is of the opinion, however, that this comparison does not obtain, inasmuch as it involves the supposition that the ratio of power of the cruising engine to that of the turbine is the same in both cases, that the percentage of full power is the same, and that the corresponding peripheral speed of the turbine will vary inversely as the square root of the total number of moving buckets.

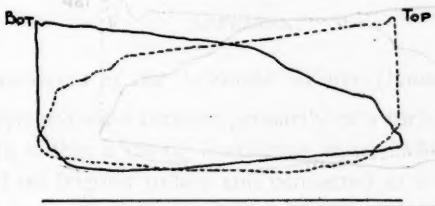
U. S. S. HENLEY

OFFICIAL SHOP TEST OF STARBOARD UNIT.

DEC. 27TH 1911. TIME 9.00 P.M.

R.P.M. 225

SCALE 150 M.E.P. TOP. 133 BOT. 133 I.H.P. 125.68



SCALE 40 M.E.P. TOP. 34.5 BOT. 31.5 I.H.P. 140.7

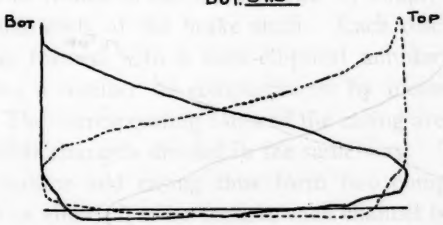


Plate VII.

A marked improvement in economy of steam consumption of the combination at cruising speeds, over that of the turbine alone, is shown in the table of comparison of results, Table II, from which it will be noted that there is an improvement of 33 per cent. at 16 knots, 62.4 per cent. at 13 knots, and 98.96 at 10 knots. These results have been reduced to standard conditions of dry saturated steam at 250 pounds gage pressure, and a vacuum of 28½ inches absolute, the method used in correcting the results being explained in the table.

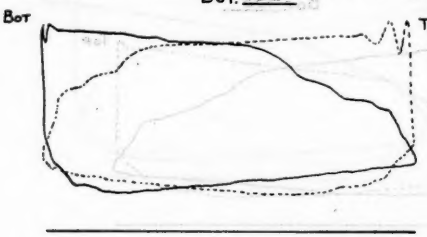
U. S. S. HENLEY

OFFICIAL SHOP TEST OF STARBOARD UNIT.

DEC. 28TH 1911. TIME 9.30 P.M

R.P.M. 290

SCALE 150 M.E.P. TOP. 153 BOT. 152 I.H.P. 185.74



SCALE 40 M.E.P. TOP. 37.5 BOT. 35.5 I.H.P. 202.17

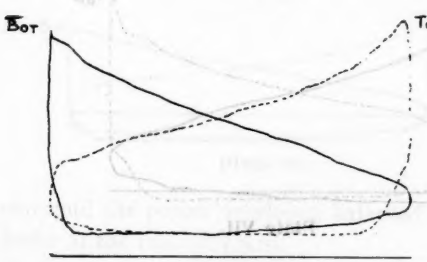


Plate VIII.

PRELIMINARY TESTS.

As originally installed the exhaust from the cruising engine was led to the fifth-stage shell of the turbine through a 6½-inch diameter pipe subdivided into two 4½-inch connections at the upper half of the turbine casing. The first three preliminary trials were run with the steam thus admitted into the fifth stage of the turbine. This connection to the turbine,

however, gave an unsatisfactory economy, and the exhaust connection was changed to the turbine ahead steam chest, and subsequently to the first-stage shell, and it is under the last condition that the official shop tests were run.

Plate V shows curves of performance of the unit, and on plates VI to VIII, inclusive, are shown some of the indicator cards obtained on the cruising engine.

APPENDIX.

Description of the "Froude" Water Dynamometer.

The dynamometer consists primarily of a turbine or rotator, revolving within a casing containing water, which is suitably mounted on friction rollers and connected to a water supply. The rotator is fixed to the shaft which projects on either side of the casing, and to which the engine or motor to be tested is coupled. Engines or motors of either direction of rotation can thus be tested on the one machine by simply coupling up to alternate ends of the brake shaft. Each disc face of the rotator is formed with a semi-elliptical annular channel divided into a number of compartments by means of oblique vanes. The corresponding faces of the casing are also formed with similar channels divided in the same way. The channels on the rotator and casing thus form two complete annular channels of elliptical cross section, each channel being divided, as stated above, into compartments by means of the oblique

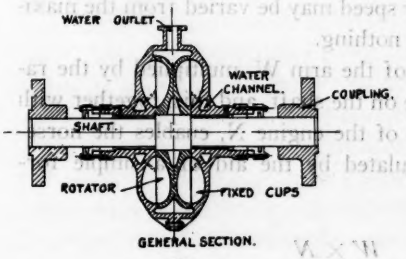


Fig. 1.

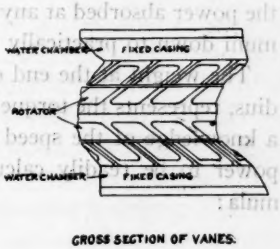


Fig. 2.

vanes. This will be understood on reference to figures Nos. 1 and 2.

When in action the water in each annular channel is rotated continuously by the centrifugal force imparted to it by the rotator, and passes from one compartment into the next, and so on. An extremely high speed of rotation of the water is obtained, and the power or energy put into the dynamometer is by this means converted into heat, which passes away in the water leaving the machine. The motion of the water causes the rotator to re-act on the casing, and tends to turn it on the friction rollers. This is prevented by means of an extension arm, working between stops, at the end of which are the balance weights by which the actual power put into the dynamometer is measured. The friction rollers supporting the casing are mounted on ball bearings, and the machine is thus very sensitively balanced, and great accuracy is assured. It will, of course, be understood that the friction of the dynamometer shaft in the bearings re-acts on the casing, and is measured at the end of the arm, and no error is introduced on this account. The friction rollers supporting the casing are adjustable, so that the dynamometer shaft may be readilv brought into line with that of the engine or motor under test.

The power absorbed at any particular speed depends to a great extent on the condition of the inlet and outlet water valves. A patent system of linkage controls these valves so as to maintain an absolutely constant resistance on the engine shaft. This machine possesses a very wide range, in fact, the power absorbed at any speed may be varied from the maximum down to practically nothing.

The weight at the end of the arm W, multiplied by the radius, represents the torque on the shaft, and this, together with a knowledge of the speed of the engine N, enables the horsepower to be readily calculated by the aid of a simple formula:

$$\frac{W \times N}{K}.$$

The length of the arm is arranged so as to give a simple constant; for example, K equals 1,000 if the arm is made 5 feet 3 inches long. For the instrument used K equals 700.

The quantity of water used by the brake is approximately 2 gallons per horsepower-hour. It depends, of course, on the temperature at which the water is allowed to run to waste. This may be between 150 degrees F. and 170 degrees F.

THE NAVAL RECIPROCATING STEAM ENGINE, ITS CHARACTERISTICS, DIMENSIONS, AND ECONOMICS.

By Ernest N. Janson, Member.

A considerable portion of technical literature has for years been devoted to marine engineering topics, among which those concerning naval machinery have been exploited extensively. Naturally, therefore, it is with some degree of reluctance that the matter is again attacked along lines designated by the above title.

The value of data and calculations pertaining to the reciprocating engine in its present state of satisfactory development, collected in a state of easy review, it is believed, will serve a useful purpose, and, it is hoped, will justify this paper.

REVIEW.

In connection with the reappearance of the reciprocating engine for propelling purposes in battleships, and as auxiliary cruising engines in recent destroyer installations, it is interesting to recall certain predictions that have been made from time to time as to its development. Thus opinions were expressed by engineering authorities more than a decade ago that pressures of 500 pounds (even 1,000) might be used; that with the turbine introduced for propelling purposes in ships the engine would disappear, and that with the rapid development of the internal-combustion engine both of the former would be displaced, all of which is yet to come.

While numerous noteworthy improvements have been made both in details of construction and workmanship, and particularly in the general arrangements of the machinery, really

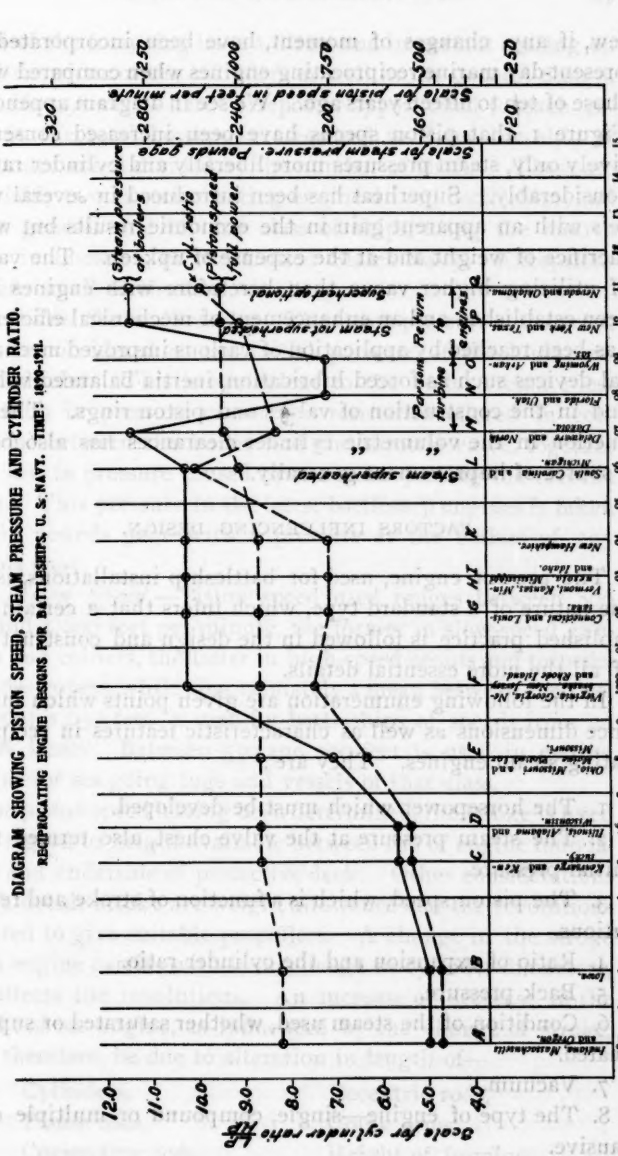


Fig. 1.

DIAGRAM SHOWING PISTON SPEED, STEAM PRESSURE AND CYLINDER RATIO.

RECIPROCATING ENGINE DESIGNS FOR BATTLESHIPS, U. S. NAVY. TIME: 1890-1911.

few, if any, changes of moment, have been incorporated in present-day marine reciprocating engines when compared with those of ten to fifteen years ago. We see in diagram appended, Figure 1, that piston speeds have been increased conservatively only, steam pressures more liberally and cylinder ratios considerably. Superheat has been introduced in several vessels with an apparent gain in the economic results but with sacrifice of weight and at the expense of upkeep. The value of utilizing higher vacua than heretofore with engines has been established and an enhancement of mechanical efficiency has been reached by application of various improved mechanical devices such as forced lubrication, inertia balanced valves and in the construction of valves and piston rings. The reduction in the volumetric cylinder clearances has also been a source of improvement generally.

FACTORS INFLUENCING DESIGN.

The present engine, used for battleship installations, is in the nature of a standard type, which infers that a certain established practice is followed in the design and construction of all the more essential details.

In the following enumeration are given points which influence dimensions as well as characteristic features in reciprocating steam engines. They are:

1. The horsepower which must be developed.
2. The steam pressure at the valve chest, also termed the initial pressure.
3. The piston speed, which is a function of stroke and revolutions.
4. Ratio of expansion and the cylinder ratio.
5. Back pressure.
6. Condition of the steam used, whether saturated or superheated.
7. Vacuum.
8. The type of engine—single, compound or multiple expansive.

All of the foregoing have a direct and important bearing on the dimensions of the cylinders as well as the general design.

1. *Horsepower.*—The horsepower apportioned to each engine is the total horsepower required to drive the ship at maximum speed divided by the number of shafts. These are invariably two in reciprocating engine-driven battleships, and is the general rule for any class of ships, using the motive power in question, in the United States Navy. Exceptions concern small or obsolete vessels.

2. *Initial Steam Pressure.*—The steam pressure at the valve chest is considered as the initial pressure at the high-pressure cylinder. This is equivalent to the boiler pressure less the drop in pressure caused by resistance offered in connections between the boiler and the throttle valve, together with loss in pressure caused by the steam passing the throttle valve. This pressure in the latest battleship engines is taken at 265 pounds gage with a pressure at the boilers of 295 pounds gage.

3. *Piston Speed.*—Piston speed used ranges between 800 feet and 1,200 feet per minute, the former in slow-speed gunboats and colliers, the latter in high-speed scouts and torpedo-boat destroyers, while approximately a mean between the two, or about 1,050 feet, is used for battleships of speeds from 20 to 20½ knots. Between 550 and 700 feet is used in engine designs for sea-going tugs and vessels of that class.

The piston speed having been determined, the stroke is fixed with regard to height available between top of engine foundations and underside of protective deck. Other considerations which decide stroke are weight allowance and the revolutions required to give suitable propellers. A change in the stroke of an engine carries with it no change in cylinder diameters, but affects the revolutions. An increase or decrease in the weight of an engine, as influenced by the change of stroke, will, therefore, be due to alteration in length of—

Cylinders,	Eccentric rods,
Piston rods,	Crank arms,
Connecting rods,	Height of framing.

Also as influenced by change in revolutions to alteration in diameter of—

Crank shafts,	Bearings,
Line and thrust shafts,	Propellers.
Propeller and stern-tube shaft,	

For a given piston speed the revolutions are indirectly proportional to the stroke.

The engines for all of the high-speed battleships are made with 48 inches stroke, the revolutions, according to speed, varying between 115 and 130 per minute at full power.

4. *Ratio of Expansion.*—The ratio of expansion and cylinder ratio are determined by the magnitude of initial pressure to be used in conjunction with the terminal pressure expected. The definition of the term "ratio of expansion" given in the Code (1902) of the American Society of Mechanical Engineers in Rules for Conducting Steam-Engine Tests is as follows:

"The ratio of expansion for a simple engine is determined by dividing the volume corresponding to the piston displacement, including clearance, by the volume of the steam at cut-off, including clearance.

"In a multiple-expansion engine it is determined by dividing the net volume of the steam indicated by the low-pressure diagram at the end of the expansion line, assumed to be continued to the end of the stroke, by the net volume of the steam at the maximum pressure during admission to the high-pressure cylinder."

The latter definition transformed into a formula may be approximately expressed:

$$\text{Ratio of expansion} = \frac{\text{L.P. capacity}}{\text{H.P. capacity at cut-off}}$$

without regard to clearance or point of release in the final cylinder. As the theoretical mean pressure depends principally upon the initial pressure together with the ratio of expansion, it is evident that, due to the fact that the percentage clearance in the low-pressure cylinder is smaller than that in the high-pressure cylinder, the actual ratio of expansion, which takes into ac-

count the clearance percentages, becomes smaller than the nominal ratio which precludes them.

However, it being not theoretical pressures which are used in figuring cylinder diameters, but actual mean effective or referred pressures as established by comparison with engines of known performance, it becomes apparent that really very small importance need be attached to whether nominal or actual ratios of expansion are being used in the calculations. But owing to the fact that "card effect" is usually ascertained from engines, the mean pressure in which is calculated upon the basis of the actual ratio of expansion, and the referred pressures always with reference to the net area of piston, total developed horsepower and piston speed, it becomes imperative, for reasons of conformity, to use actual ratios of expansion in calculations.

Expressions for the ratio of expansion and cylinder ratio in multiple-expansion engines are:

$$\text{Nominal rate of expansion} = \frac{2A_L - d}{(2A_H - d)H_c};$$

$$\text{Actual ratio of expansion} = \frac{(2A_L - d)(1 + c)}{(2A_H - d)(H_c + c_1)};$$

$$\text{Cylinder ratio } r_H = \frac{2A_L - d}{2A_H - d};$$

where,

A_L = Low-pressure cylinder area;
A_H = High-pressure cylinder area;
d = Piston-rod area;
H_c = High-pressure cylinder cut-off;
c = Clearance percentage, L.P. cylinder;
c_1 = Clearance percentage, H.P. cylinder.

With increased ratio of expansion the weight of the engine is increased and cylinder condensation augmented, the latter being due not only to greater adiabatic condensation, but also to the increased surface without correspondingly added steam

volume at admission. Mechanical efficiency is also impaired as a result of greater friction with the enlarged surface.

Economy at full power with naval vessels being of less importance than when operating under reduced or cruising conditions, engines with moderate ratios of expansion will meet service conditions more satisfactorily than those having these ratios unduly increased. In the very latest designs of 4-cylinder, triple-expansion engines, with two low-pressure cylinders, the receiver pipe forming the connection from the I.P. cylinder valve chest to each low-pressure valve chest is fitted with a gate valve by means of which either low-pressure cylinder may be cut out. This will give more nearly the proper ratio of expansion at reduced power than would be the case with both cylinders in operation.

5. *Back Pressure.*—Back pressure, P_B, is the pressure of cylinder exhaust. This is a function of the pressure at release, P_r, and the condenser pressure, P_c. It may be taken approximately as a mean of these two pressures, thus:

$$P_B = .5\,(P_r + P_c).$$

The principal causes for augmentation in the back pressure over that given by the foregoing expression is resistance in the valve ports, density of the steam, the velocity with which the piston moves and the presence of air in the steam.

The average back pressure in naval engines is about 6 pounds absolute. This pressure, however, in the *Delaware*, when running at full power and 128 r.p.m., with about 26.3 inches of vacuum, was only $4\frac{1}{2}$ pounds, and at 87 r.p.m. not exceeding $1\frac{1}{2}$ pounds. In the U. S. S. *Georgia*, on the contrary, on a full-power run, it amounted to as much as $12\frac{1}{2}$ pounds.

While vacuum aids in decreasing the back pressure it does so only nominally, and must of necessity always be better in the condenser than in the cylinder by an appreciable amount in order to maintain the flow of exhaust at a certain velocity.

In Figure 2 is illustrated the pressures existing at the exhaust end of the low-pressure cylinder.

P_r (pressure at point of release) is not that due alone to the steam expansion in the engine, but is much below, owing to various causes of loss, such as cylinder condensation and pressure drop in the receivers, etc. Neither is the pressure of the condenser, P_c, due to the pressure of condensation within the condenser, but also to that caused by the presence of air in the exhaust steam.

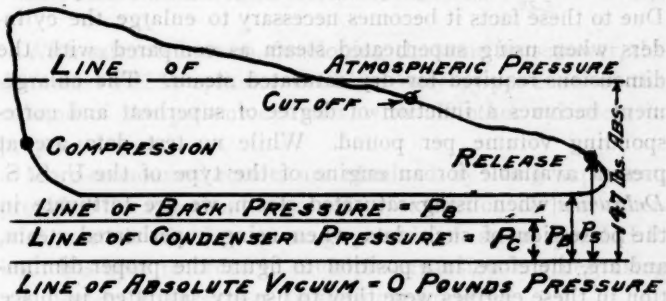


Fig. 2.

6. *Condition of Steam.*—The quality of the steam has an important bearing upon the economic performance of engines. From tests made on different types of battleship boilers used for generation of steam without superheat up to date it may be inferred that the steam is practically dry. While only temporary means have been devised during trials of ships to ascertain the quality of the steam, it is believed that the dryness fraction is not below 98 to 99 per cent. in actual service. Superheated steam has been used in some of the later ships. The *South Carolina* and *Michigan*, the *Delaware* and the *North Dakota* had superheaters fitted on all boilers, giving an average of about 75 degrees F. superheat, the *Texas* and *New York* have eight out of twelve boilers arranged with superheaters, while in the two latest ships, the *Oklahoma* and *Nevada*, the fitting of superheaters is not required.

The use of superheated steam raises the efficiency of the fluid by adding heat without increasing the pressure, lessens or prevents condensation during expansion and reduces the

resistance to flow and steam friction in the cylinder ports on account of the diminished density of the steam.

The specific volume of superheated steam, in cubic feet per pound, is greater than that of dry saturated steam at the same steam pressure, and the difference increases with the degree of superheat. The total heat of the former is also considerably greater than of the latter for the same pressure conditions and increases in proportion to the degree of superheat. Due to these facts it becomes necessary to enlarge the cylinders when using superheated steam as compared with the dimensions required for dry saturated steam. The enlargement becomes a function of degree of superheat and corresponding volume per pound. While no test data are at present available for an engine of the type of the U. S. S. *Delaware* when using saturated steam, we are fortunate in the possession of such data when using superheated steam, and are, therefore, in a position to figure the proper diminution in these engines were they to use dry saturated in place of superheated steam, thus:

At maximum trial performance steam consumption per I.H.P. hour, pounds	13.38
Average superheat at engine, degrees F.	61
Steam pressure at valve chest, pounds absolute	268
Temperature corresponding, degrees F.	467
Specific volume of steam, in cubic feet per pound	1.92
Total heat per pound of steam, B.T.U.	1,243.1
Temperature of feed water, degrees F.	147
B.T.U. in feed water above 32 degrees F., per pound	115
chargeable to engine per pound of steam = 1,243.1 — 115	1,128.1
per I.H.P. hour = 13.38 × 1,128.1 =	15,095
High-pressure cylinder, displacement at cut-off, in cubic feet per I.H.P. hour = 13.38 × 1.92 =	25.7

For dry saturated steam we have, assuming same steam pressure, or 268 pounds absolute and 1 per cent moisture in the steam:

Temperature corresponding, degrees F.	407.1
Specific volume, in cubic feet per pound	1.73
Heat of evaporation corresponding to 268 pounds absolute pressure	824.9
Reduced to 1 per cent. moisture = .99 × 824.9	816.65

Heat in liquid at 268 pounds absolute	381.2
Total heat corresponding, per pound	1,197.85
B.T.U. in feed at 147 degrees F. temperature	115
chargeable to engine per pound of steam = 1197.85 − 115	1,082.85
Theoretical steam consumption per I.H.P. hour, pounds $= \frac{15,095}{1082.85} =$	13.94
High-pressure cylinder displacement, at same cut-off, in cubic feet per I.H.P. hour $= 13.94 \times 1.73 =$	24.11
Ratio of H.P. cylinder when using saturated and superheated steam under conditions stated above $= \frac{24.11}{25.7} =$	0.93

With saturated steam, developing the same horsepower, the *Delaware's* H.P. cylinder needs be only 93 per cent. of the volume of the present cylinder, with all other cylinders in proper relation.

More economical results are obtained by using superheat, provided the superheat is gained by extraction of heat otherwise wasted. As low as 10.5 pounds of steam per I.H.P. hour has been recorded in multiple-expansion engines, using 300 degrees F. superheat, in stationary practice.

7. *Vacuum.*—Under the heading "back pressure," reference has been made to the influence of vacuum. While an increase in the vacuum augments the power of the engine, other conditions remaining equal, the overall economic results are not always improved. With triple or quadruple-expansion engines, the general economy, with different degrees of vacuum, is largely dependent on the proportion of auxiliary steam used by the installation, and whether this auxiliary steam is being used for feed-water heating. If the auxiliary steam constitutes a large percentage of the main steam used, a high feed temperature may be obtained even though the vacuum is considerably above that ordinarily used with reciprocating engines. This is, of course, due to the comparatively large heat transference from the auxiliary steam to the air-pump discharge through the feed heater. While it is quite possible and comparatively easy, provided there is a sufficiency of condensing surface and an adequate amount of comparatively cold injection water, to produce a very high vacuum in

Table 1.—TABLE OF PRESSURES, RATIOS AND CONSTANTS FOR SOME RECENT SHIPS IN U. S. NAVY.

Name of ship.	One engine. I.H.P.	Pressure per sq. in. lbs. abs. P_1	P_0	R_1	$\frac{A_L}{A_H}$	$\frac{P_0}{R_1}$	Saturated or superheated.	Steam in receivers.	Pounds per square inch. Abs. $P_M = P_1\frac{1+\log_e R_1}{R_1}$	P_R	Abs. P_B	$P_M - P_B = P_E$	Inches by gage. P_V	P_C	$\frac{P_E}{P_E} = F$
BATTLESHIPS AND A. CRUISERS.	Indicated horsepower.	H.P. cyl. valve ch. pr.	H.P. cyl. cutoff pres. (cards).	Actual ratio of exp.	Cylinder ratio.	Term. pressure.			Theoretical mean pressure.	Referred pressure.	Back pressure.	Theoretical mean effective pressure.	Vacuum in L.P. cyl.	Vacuum in condenser.	"Card effect" mean eff. pressure constant.
South Carolina..	8,225	256.0	150.0	12.1	10.36	12.5	47.0° sup.	No.	256.0×.288 =73.9	37.04	3.5	73.9—3.5 =70.4	22.78	26.24	.526
	1,860	131.3	...	14.08	10.36	...	Sat.	No.	131.3×.265 =33.79	12.53	1.51	33.79—1.51=32.28	26.84	26.84	.382
Michigan.....	8,008	260.0	220.0	12.6	10.36	17.7	85.6° sup.	No.	260.0×.2807=72.98	34.23	3.16	72.98—3.16=69.82	23.47	26.97	.493
	1,745	88.0	...	12.3	10.36	...	51.0° sup.	No.	88.0×.285 =25.12	12.35	1.66	25.12—1.66=23.46	26.53	27.35	.527
Delaware.....	14,289	268.0	220.0	8.71	7.92	25.2	61.6° sup.	No.	268.0×.364 =97.55	50.9	4.46	97.55—4.46=93.09	20.82	26.3	.545
	1,753	82.0	...	8.71	7.92	...	36.2° sup.	No.	82.0×.364 =32.03	13.34	1.41	32.03—1.41=30.5	26.94	27.97	.437
North Carolina..	13,019	258.0	234.7	8.38	7.5	Indef.	Sat.	Yes	258.0×.373 =96.23	53.2	9.67	96.23—9.67=86.56	10.21	25.1	.61
	8,900	216.7	...	8.54	7.5	...	Sat.	Yes.	216.7×.368 =79.75	45.4	...	.....	...	25.4	...
Birmingham.....	7,738	240.0	241.0	11.5	10.8	20.8	Sat.	No.	240.0×.30 =72.0	36.3	5.5	72.0 —5.5 =66.5	18.72	26.5	.546
	798	64.0	...	11.5	10.8	...	Sat.	No.	64.0×.30 =19.2	8.15	1.52	19.2 —1.52=17.68	26.8	26.8	.304
Mississippi.....	6,862	259.0	216.0	9.0	7.5	Indef.	Sat.	Yes.	259.0×.355 =91.9	60.49	7.3	91.9 —7.3 =84.6	15.04	26.0	.64
	...	...	...	...	...	...	...	...	.....	...	...	.....	...	...	...
COLLIERS.															
Cyclops.....	3,368	192.0	138.0	12.04	7.63	11.5	Sat.	No.	192.0×.2903=55.75	33.75	2.58	55.75—2.58=53.17	24.66	27.0	.634
	...	...	...	...	...	...	...	...	.....	...	...	.....	...	...	...
Hector.....	1,960	195.0	168.7	10.12	7.7	16.6	Sat.	No.	195.0×.3277=63.9	35.01	5.4	63.9 —5.4 =58.5	18.9	25.84	.601
	...	...	...	...	...	...	...	...	.....	...	...	.....	...	...	...

the condenser, this is by no means so as ragards the cylinder. In order for the vacuum to have pronounced influence upon the performance of the engine, it must exist in the cylinder, without which not even the highest vacuum in the condenser will do much good. A difference of several inches in the vacuum of the condenser and that of the cylinder is the general rule in naval engines, see Table I. This is due in the first place to incomplete expansion as a result of a comparatively small expansion ratio, and secondly to an increase in back pressure from causes previously cited. Late tests undertaken by Belliss, Morcom & Company, Ltd., have shown that while the saving in steam consumption at 27½ inches of vacuum, or thereabouts, over that at 21½ inches is considerable, the saving is only nominal between 25½ and 27½ inches. Figures of actual tests are given in the table below (from paper read by D. B. Morison, before the North-East Coast Institution of Engineers and Shipbuilders, England).

Size of engine (18½ inches + 27 inches + 40 inches) 20 inches stroke; B.H.P. = 1,080 constant for four tests; superheated steam used.

Vacuum, in inches	27.9	26.8	25.8	21.5
Steam consumption, in pounds per B.H.P..	12.7	13.2	12.83	14.16
When reduced to 100 degrees F., superheat..	12.55	12.68	12.74	14.15

To increase the vacuum involves changes in various auxiliaries as well as the condenser itself. First, a larger and more efficient condenser must be produced; secondly, an increase in air and circulating-pump capacity must be provided, and means must be adopted to effectively prevent air leakage.

While it is true that very little is gained in the steam economy, as shown in the above test, and an actual loss may be incurred if such arrangements as heating the feed by auxiliary exhaust be not resorted to, raising the vacuum does increase the power of the engine.

In Table 1 are given corresponding absolute pressures, vacuum, in inches of mercury, and temperatures, in degrees.

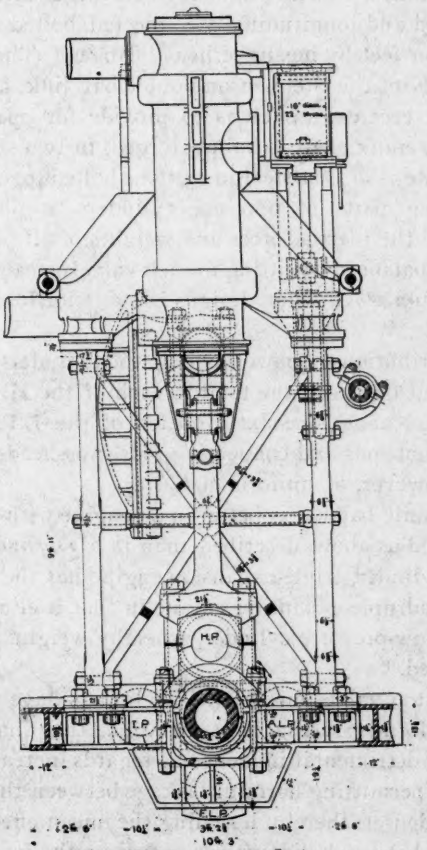
Table 2.—Temperatures and Pressures of Saturated Steam.

Vacuum, inches of mercury when barometer is at 30 ins.	Absolute pressure, pounds per square inch.	Temperature, degrees F.
29.8	0.0982	36
29.7	0.1474	46
29.6	0.1965	53.5
29.5	0.2456	60
29.4	0.2947	65
29.3	0.3438	69
29.2	0.3929	73
29.1	0.4421	76
29.0	0.4912	79.5
28.9	0.5403	82.5
28.8	0.5894	85
28.7	0.6385	87.5
28.6	0.6876	90
28.5	0.7368	92
28.4	0.7859	94
28.3	0.8350	96
28.2	0.8841	98
28.1	0.9332	99.5
28.0	0.9823	101.5
27.9	1.031	103
27.8	1.081	104.5
27.7	1.130	106
27.6	1.179	107.5
27.5	1.228	109
27.4	1.277	110.5
27.3	1.326	111.5
27.2	1.375	113
27.1	1.424	114
27.0	1.474	115
26.9	1.523	116.5
26.8	1.572	117.5
26.7	1.621	118.5
26.6	1.670	119.5
26.5	1.719	121
26.4	1.768	122
26.3	1.817	123
26.2	1.866	124
26.1	1.916	125
26.0	1.965	126
25.8	2.063	128
25.6	2.161	129
25.4	2.259	131
25.2	2.358	133
25.0	2.456	134
24.8	2.554	135.5
24.6	2.652	137
24.4	2.751	138
24.2	2.849	139.5

8. *Type of Engine.*—The type of engine generally used in battleship installations of the United States Navy is the 4-cylinder, triple-expansion engine. This general design has

been used iu some twenty ships of late design, which incorporates the general constructive features of the *Delaware's* engines, shown on Figures 3 and 3-A. Engines previous to those of the *Delaware*, however, were not equipped with forced lubrication, and only in two instances, in the *South Carolina* and *Michigan*, used superheated steam.

The general arrangement of cylinders is: Forward L.P.



H.P. Cylinder looking Ford.

Fig. 3A.

cylinder, H.P., I.P. and after L.P., with the H.P. cylinder leading, followed by the I.P., forward L.P. and after L.P., in the order named. The H.P. and forward L.P. are opposite and 180 degrees apart, which is also the case with the I.P. and after L.P., each pair of cranks being 90 degrees apart. All cylinder valves are of the piston type, one for the H.P. cylinder and two for each of the other cylinders. The framing is built up of turned and finished forged-steel columns, cross braced and longitudinally connected, bolted to bed plate and cylinder feet by means of heavy flanges. The valve gear is invariably of the Stephenson double-bar link, and the setting of the eccentrics such as to provide for open eccentric rods. The crank shaft is hollow forged, in two sections, and the bed plate is of cast steel in sections bolted together. The reciprocating parts of opposite cylinders are balanced in weight, and the inertia forces and weights of all piston valves are counterbalanced by fitting to each valve a steam-cushioned balance piston. All later designs provide for forced lubrication.

The distribution of power among the cylinders is such as to give about one-third the total to each of the H.P. and I.P. cylinders and about one-sixth to each of the L.P. cylinders. Guides, piston rods and connecting rods, crossheads and crank pins are, however, of uniform size.

The dynamic balance is better in an engine with four cylinders arranged as above described than in a 3-cylinder engine, and the 4-cylinder, triple-expansion engine has the advantage over the quadruple-expansion engine in that it eliminates the very large low-pressure cylinder, whereby weight as well as space is saved.

While piston valves on the L.P. cylinders have the advantage of uniformly satisfactory operation, they have at the same time a detrimental influence as regards increasing clearance and in permitting harmful leakage between the cylinder and the condenser, thereby lessening the mean effective pressure. Well balanced slide valves overcome the foregoing defects and can be made double or treble ported. This greatly

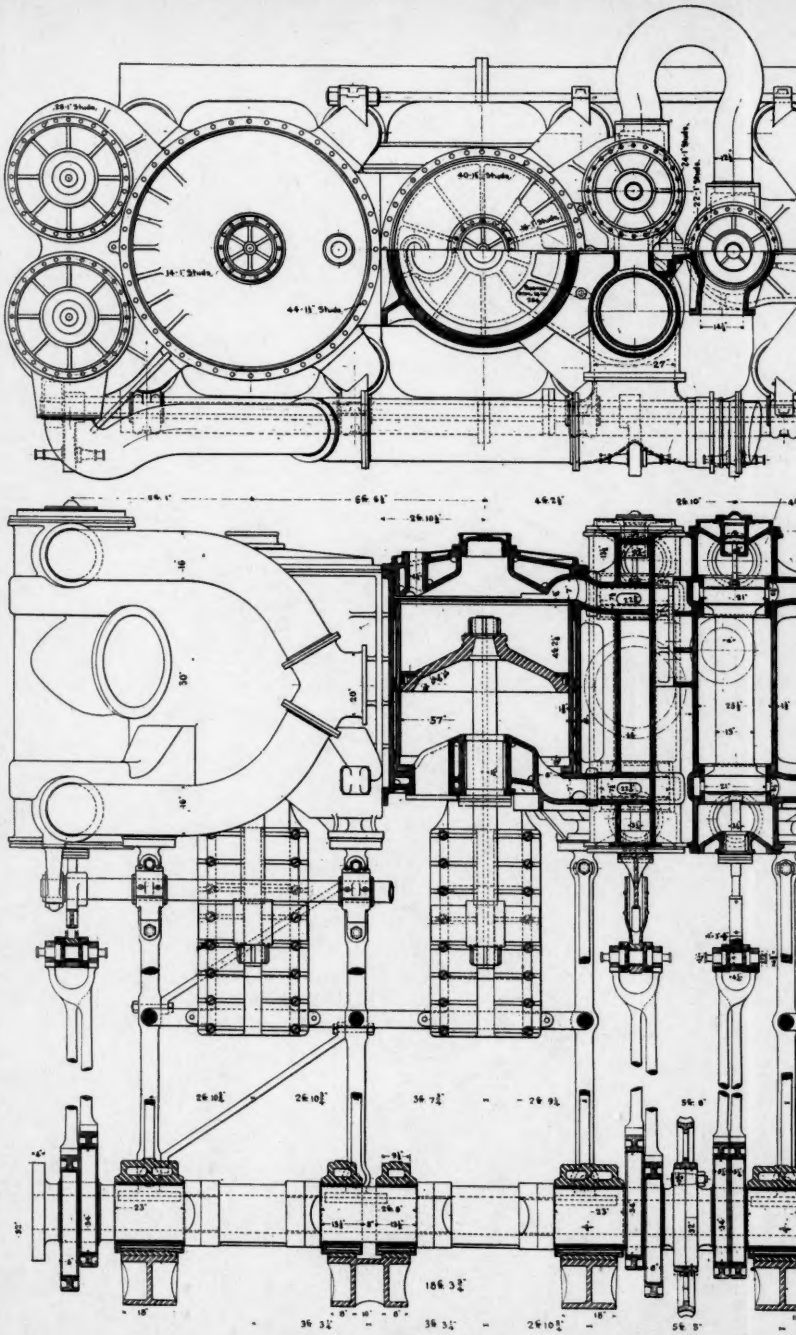
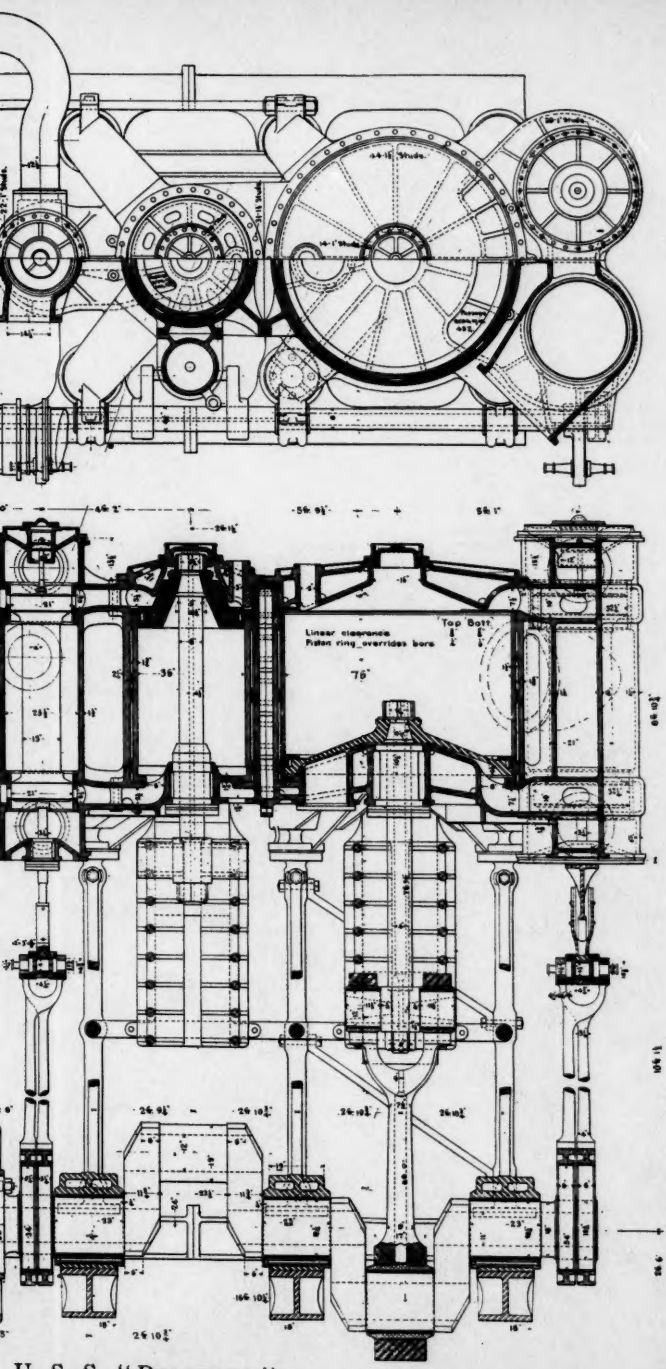


Fig. 3.—Starboard Engine, U. S. S.



U. S. S. "DELAWARE."

increases the steam and exhaust openings, giving a less obstructed flow, which increases the mean pressure and, therefore, the power of the engine.

H.P. and I.P. cylinder piston rings are made solid, but the L.P. cylinder piston rings are split and provided with means of obtaining tightness. All valve-packing rings are made with a solidly bolted joint admitting of adjustment for wear.

SOURCES OF LOSS.

That which stands out most prominently to the engineer, when an analysis is made of the results obtained with engines working under certain given conditions, are the losses encountered. These may be ascribed principally to the following causes:

1. Wire drawing, caused by slow, retarded and inadequate steam and exhaust openings in the cylinder valves.

2. Leakage in cylinder valves and pistons and steam friction in the ports.

3. Drop in pressure in steam connections and receivers.

4. Incomplete expansion—expansion ceases at the point of release instead of the end of the stroke of the last cylinder.

5. Linear and volumetric cylinder clearances. The latter increase with the size of cylinder ports, and are larger with piston valves than with flat valves.

6. Cylinder condensation, which is the source of greatest thermodynamic loss. This is caused largely by the alternate heating and cooling of the cylinder walls and the great difference existing in the temperature of steam admission and outlet. Cylinder condensation may be reduced by minimizing the temperature drop between the several cylinders. This is done by making the engine of the multiple-expansive type, and by providing steam jacketing, superheat and high compression.

7. Mechanical inefficiency as caused by friction in moving parts, stuffing boxes and bearings. The friction in a marine engine is greater than in a stationary engine, due to the long

line shafting and to the possible working of the ship in a seaway. It consists of:

(*a*) Dead-load friction, directly proportional to revolutions.

(*b*) Friction of the working load, which varies directly with mean effective pressure and, therefore, the power developed.

Improved methods in design and construction, together with forced lubrication, assistant cylinders on the main valves, and more perfect balance has reduced friction to such a degree that the loss through this source at full load should not exceed 6 to 8 per cent. of the total power. The dead-load friction, or the resistance within the engine when turning over with no steam on, being practically constant, constitutes a larger proportion of the power at reduced speed than at full speed. Owing to this fact the mechanical efficiency at reduced power falls much below that at full power and, probably, does not exceed 60 or 70 per cent.

FACTORS CONDUCIVE TO ECONOMY.

The propelling engines for naval vessels differ materially with respect to economy from those of the merchant marine. The best economy of the latter is sought at full power or nearly so, while with the former a reasonably good economy is also expected at much reduced speeds. The means which tend to improve economic results are the following:

1. *The Use of the Highest Practical Steam Pressure.*—By increasing the steam pressure the temperature is also increased. High temperatures have, however, been found to destructively affect certain materials like copper and certain compositions containing this metal, for which reason principally it is necessary to use wrought and cast steels in construction of high-pressure steam lines. Other causes of trouble with very high steam pressures are the greater amount of expansion taking place in the piping, and the necessity for its effectual compensation, as well as in the greater difficulty of producing efficient joints.

While practical difficuties arise from high steam pressures,

thermodynamic gains ensue to a marked degree, since from Carnot's Cycle the thermodynamic efficiency is measured by the ratio of the difference of the absolute temperature at reception and rejection and that at reception.

Thus $\frac{T_1 - T_2}{T_1}$ represents the efficiency of a perfect heat engine. The greater the initial temperature the smaller is condensation resulting from expansion. In this connection it has been pointed out by Rankine that for temperatures in the vicinity of 1,000 degrees F. there is no condensation during expansion, while for still higher temperatures superheating would actually occur.

2. *High Piston Speed.*—To increase the piston speed adds neither to the cost nor the size of the engine, but adds in a direct ratio to the power. Thus the same engine with 1,000 feet piston velocity develops twice the power of the one with only 500 feet, and the former is thermodynamically the more efficient because it affords less time for condensation by the cylinder walls. To increase the piston speed considerably, beyond that presently used, is being looked upon as a possible means of greatly improving the utility of the reciprocating engine as a prime power.

Advantages of high speed in condensing engines, as regards effect on cylinder condensation, have been shown by tests made by the late Mr. Willans, results of which are shown in the table below.

Revolutions per minute	401	301	198	116
Percentage of water present at H.P. cylinder cut-off	8.9	12.2	17.9	20.9
Steam consumption per hour per I.H.P., in lbs.	17.3	17.6	18.9	20

3. *Superheat.*—The effect of superheat upon the performance, size and efficiency of engines has been previously alluded to.

4. *Steam Jackets.*—Professor Robert H. Thurston, in an article in the Journal of the Franklin Institute in 1891, stated: "The fact is sufficiently well known that the steam jacket as

employed on the steam engine, of whatever form and arrangement, is intrinsically a wasteful element, and that its use only gives in certain cases an economical advantage by its repression of wastes of large magnitude. It checks wastes more or less effectively and completely." It is, however, generally conceded that a steam jacket on the low-pressure cylinder reduces cylinder condensation and accelerates the process of reëvaporation. It prevents, also, to some degree, the action upon the steam from the alternate cooling and heating of the cylinder walls.

5. *Forced Lubrication.*—This reduces friction of moving parts, prevents heating, and generally improves the mechanical efficiency.

6. *Balance Pistons.*—As now designed their application conduces to reduce strains in the valve gear and to minimize friction and vibration.

7. *Small Cylinder Clearances.*—It being somewhat difficult, with the present arrangements of valve gears, to bring up to initial pressures the compression, a considerable waste of steam is caused by the filling with fresh steam the clearance spaces of the cylinders. By making the ports shorter and more direct and by using higher velocities this source of loss may be greatly reduced.

8. *Tightness of Pistons and Valves.*—The preventing of leakage of steam from one end of the cylinder to the other, or into receivers and condensers, has an important effect on the mean effective pressure, upon which the power of the engine largely depends.

DETERMINATION OF CYLINDER DIAMETERS.

It is important so to proportion the diameters of the cylinders in a multiple-expansion engine as to make each cylinder develop, as nearly as possible, an equal amount of power. Having succeeded in doing this it may be assumed that the working parts of the respective cylinders, such as the slipper, crosshead pins, crank pin and main bearings will undergo practically equal amounts of wear throughout the engine.

While such proportions are quite readily obtained with the 3-cylinder triple and 4-cylinder quadruple-expansion engines, it is not a point aimed at in 4-cylinder triple-expansion engines, such as are commonly used for battleship installations. In those engines each low-pressure cylinder does not, even nearly, develop the same horsepower as either the high or intermediate-pressure cylinders. But owing to primarily liberal allowances in bearing surfaces and, secondarily, to the practical advantages existing in maintaining uniform dimensions, the working parts are of equal dimensions for all four cylinders. By this means, also, a more satisfactory balance of the moving parts is accomplished.

Different methods are used in calculating cylinder diameters of multiple-expansive engines. Those most practical in their application are:

1. L.P. cylinder referred-pressure method.
2. H.P. cylinder volume from approximate steam consumption.
3. By comparison of power, pressure and revolutions with known engines.
4. By determination of the cylinder volumes for such temperature drops as would, approximately, divide equally the power.

Method 1.—This method seems most applicable in that the results derived from its use are dependent on nearly all the conditions which, in a greater or less degree, influence dimensions. It is based upon the supposition that the whole power expected to be developed by all the cylinders in the engine, for instance a multiple-expansive engine having one H.P., one I.P. and two L.P. cylinders, is developed by the L.P. cylinders alone, acted upon by a mean effective pressure due to given initial pressure, ratio of expansion, back pressure and all ensuing losses. We may assume one L.P. cylinder in area equivalent to the two spoken of. The pressure in question is the *equivalent mean effective pressure* or the *pressure referred to the L.P. cylinder.* This is, of course, not the pressure prevailing in the L.P. cylinder when in actual operation, as this

pressure is only about one-third of the referred pressure and, therefore, the power of the cylinder only about *one-third* of the total. The mean effective pressure of H.P. and I.P. cylinders must also be such that each develops about one-third of the total power. This will be obtained by proper cylinder ratios.

Expansion is assumed to be hyperbolic, the curve for the expansion pressure coinciding with that of the common hyperbola. The cylinder ratio is stipulated with a view to giving the most economical engine for the service intended, as well as with regard to weight and space. The cut-off in the H.P. cylinder is determined in a way such that the "ratio of expansion" gives a certain desired terminal pressure.

The initial pressure is less than the boiler pressure and may be considered to be about 30 pounds below this with machinery arrangements used in types similar to the latest battleship designs.

The back pressure is reasonably well known. As stated before, it is a function of release pressure and that in the condenser. Clearance spaces are approximately determined by comparison with similar engines.

Having carefully considered what value to give to each of the foregoing items, the procedure of calculating the cylinders may now be proceeded with. The following notations are used:

A_L = L.P. cylinder area (one or two cylinders);
A_I = I.P. cylinder area;
A_H = H.P. cylinder area;
d = Piston-rod area;
c = L.P. cylinder clearance;
c_1 = H.P. cylinder clearance;
H_c = H.P. cylinder cut-off;
I.H.P. = Indicated horsepower;
S = Piston speed in feet per minute;
L = Length of stroke in feet;
N = Number of single strokes per minute;

P_1 = Valve-chest pressure in pounds absolute;
P_M = Theoretical mean pressure in pounds absolute;
P_E = Theoretical mean effective pressure;
P_R = Actual mean effective pressure; also called referred pressure, or, equivalent mean effective pressure (P_Q);
P_B = Back pressure in pounds absolute;
P_T = Terminal pressure in pounds absolute;
r_H = Cylinder ratio, or the ratio of L.P. to H.P.;
r_I = Ratio of L.P. to I.P.;
R = Nominal ratio of expansion;
R_1 = Actual ratio of expansion;
F = Card effect;
P_o = Cut-off pressure in H.P. cylinder.

Formulæ:

$$F = \frac{P_R}{P_M - P_B} = \frac{P_R}{P_E}.$$

$$P_R = F \times P_E.$$

For F and other quantities see Table 1, which is figured for some of the ships in the naval service.

$P_T = \frac{P_o}{R}$, which has only theoretical value and is obtained direct from the card by an assumed extension of the expansion curve. The point of intersection between this extension and the normal drawn at end of stroke indicates the terminal pressure. The actual terminal pressure is greatly below the theoretical, owing to the losses sustained by condensation, drop in receivers, friction and radiation.

$$r_H = \frac{A_L}{A_H}.$$

$$r_I = \frac{A_L}{A_I} = \frac{A_L}{K\sqrt{r_H}}.$$

The $\sqrt{r_H}$ is multiplied by a constant, K, ranging in size from 1.1 to 1.15, owing to the fact that, generally, without

this multiplier the I.P. cylinder becomes too small in developing its share of the work.

$$P_M = P_1 \times \frac{1 + \text{hyp. log. } R_1}{R_1};$$

$$P_E = P_M - P_B;$$

$$A_L = \frac{\text{I.H.P} \times 33{,}000}{P_E \times S} = \frac{\text{I.H.P.} \times 33{,}000}{F \times P_E \times S};$$

$$A_H = \frac{A_L}{r_H};$$

$$A_I = \frac{A_L}{r_I}.$$

The diameters having been determined with piston speed stipulated, the stroke and revolutions are established with regard to propellers, height and weight of engine.

Method 2.—This method is based upon the fact that a steam engine requires a quantity of steam, the heat in which is sufficient not only to develop the power, but also for thermodynamic losses.

As the steam weight, or its equivalent in volume used per I.H.P., depends entirely upon the inherent features of the engine, both as regards its mechanical construction and its capacity for utilizing the steam to the best thermodynamic advantage, it is readily understood that each type of engine has a definite rating as regards the heat units required to give a stipulated return in mechanical work.

To render this method practicable, therefore, it is quite essential to possess complete knowledge of steam consumption per unit of power of the engine in contemplation and then to design it in conformity with the engine by which the comparison is made. With present practice of ascertaining steam economy of naval engines, namely, by actual water measurement during trials, this knowledge is readily obtained and, therefore, the method is rendered applicable with a certain degree of reliability.

The application of the method consists in determining the volume of the H.P. cylinder, instead of, as by Method 1, the area of the low-pressure cylinder. This is done with respect to cut-off, initial steam pressure, piston speed, power to be developed and the steam weight necessary to give the power. The steam volume per unit of weight as influenced by its quality or, in other words, whether saturated or superheated steam is being used, must be noted.

It is presumed that all of the steam necessary for developing the whole power enters the high-pressure cylinder and that no part of the power has been developed by admitting live steam into any other part of the engine, as, for instance, the receivers. It is also assumed that work due to expansion in succeeding cylinders is obtained in the ordinary manner of predetermination of ratio of expansion, with due regard to back pressure and vacuum.

The foregoing converted into a formula, giving the relations of cylinder diameter to that of power and terms influencing the power, is given below. Observing the following notations:

1. I.H.P. = Indicated horsepower;
2. P_1 = Initial steam pressure at valve chest;
3. H_c = High-pressure cylinder cut-off;
4. S = Piston speed;
5. C = Steam consumption, in pounds per I.H.P. hour;
6. A_H = Area H.P. cylinder, in square inches (exclusive of rod);
7. r_H = Cylinder ratio L.P. to H.P.;
8. r_I = Cylinder ratio L.P. to I.P.;
9. t = Temperature of steam at P_1;
10. d = Density of steam at t and P_1;

we have

$$\frac{A_H}{144} \times S \times 60 \times H_c \times d = C \times \text{I.H.P.};$$

or

$$\frac{A_H}{144} \times S \times H_e = \frac{C}{60 \times d} \times \text{I.H.P.}$$

$\frac{C}{60 \times d}$ is a constant for each type of engine with P_1 and t given.

If we put $\frac{C}{60 \times d} = K$, we have

$$A_H = \frac{K \times \text{I.H.P.} \times 144}{S \times H_e}.$$

Table 3 gives K for saturated steam, and for steam with 60 degrees F. of superheat and a steam consumption per horsepower hour, respectively, of 17.4 and 13.4 pounds, as obtained by actual tests and trial data of U. S. S. *Birmingham* and *Delaware*. Similar tables may be worked up for any other condition. The L.P. and I.P. cylinders are found in the ordinary way, observing that the ratios r_H and r_I must conform with those of the engines from which K has been derived.

Method 3 is only used as an approximation for preliminary work. Its disadvantage lies in the necessity of having access to an engine, by which comparison is made, identically similar to that to be determined. Not only the working conditions, but also the essential features of construction, must be the same. Thus, if live steam has been used in one it must be presumed to be used in the other; the condition and quality of the steam, together with arrangements of steam jackets, general design features in valves and steam ports and lagging, ratio of expansion, etc., must rigidly apply to one as well as to the other.

These items being in close conformity, in the known engine, with corresponding ones of the engine to be determined, the areas of the high-pressure cylinders of the two engines vary:

Table 3.—CONSTANTS "K" FOR DETERMINATION OF H.P. CYLINDER VOLUME BASED ON STEAM CONSUMPTION.

Pressures.		Saturated Steam. Steam consumption = 17.4 pounds, I.H.P. hour. Based on U. S. S. *Birmingham's* performance.				Superheated Steam -- 60 degrees F. Steam consumption = 13.4 pounds, I.H.P. hour. Based on U. S. S. *Delaware's* performance.			
Gage.	Absolute.	Specific volume of steam.	Density of steam = *d*.	Total heat.	H.P. cylinder constant = *K*.	Specific volume of steam.	Density of steam.	Total heat.	H.P. cylinder constant = *K*.
145.3	160	2.833	.353	1,192.8	.816	3.12	.320	1,230	.697
160.3	175	2.603	.384	1,195.0	.736	2.87	.348	1,232.2	.641
175.3	190	2.408	.415	1,197.1	.682	2.66	.375	1,234.3	.594
190.3	205	2.241	.446	1,199.0	.634	2.47	.404	1,236.1	.552
205.3	220	2.096	.477	1,200.8	.592	2.31	.432	1,237.9	.516
220.3	235	1.968	.508	1,202.6	.557	2.17	.460	1,239.6	.485
235.3	250	1.854	.539	1,204.2	.524	2.05	.487	1,241.3	.458
250.3	265	1.753	.570	1,205.8	.496	1.44	.515	1,242.8	.433
265.3	280	1.662	.602	1,207.3	.469	1.84	.544	1,244.3	.410
280.3	295	1.580	.633	1,208.8	.447	.....	.....	.....	.....

Remarks: "Constants" based on developed I.H.P. and actual steam consumption obtained by weighing water. Steam consumption assumed constant for range of pressures given, being based on ratio of expansion such as to give same terminal and referred pressure throughout the whole range of pressures.

Constant "K" figured from $K = \frac{\text{Steam per I.H.P. hour.}}{60 \times \text{density}}$

1. Directly as the horsepower;
2. Inversely as the piston speed;
3. Inversely as the steam pressure.

If

A_K = area of H.P. cylinder in the known engine;
I.H.P.$_K$ = indicated horsepower in the known engine;
S_K = Piston speed in the known engine;
P_K = Steam pressure in the known engine;

and

A_s, I.H.P.$_s$, S_s and P_s denote, respectively, the same quantities in the engine sought, we have

$$A_s = A_K \frac{\text{I.H.P.}_s}{\text{I.H.P.}_K} \times \frac{P_K}{P_s} \times \frac{S_K}{S_s}.$$

Having found the area of the H.P. cylinder, those of the I.P. and L.P. are found by multiplying by ratios chosen for I.P. to H.P. and L.P. to H.P., respectively.

Method 4 may be termed the thermal method of determining cylinder volumes, because of its treating directly with steam temperatures as existing in the different cylinders of a multiple-expansion engine and the steam volumes corresponding thereto.

This method is not used with any degree of accuracy and is seldom applied for practical use, owing to the difficulty of so apportioning to each cylinder the temperature range which will give to it its proportional share of work. This difficulty arises from want of knowledge in accurately determining the heat consumed in overcoming losses due to cylinder condensation, radiation, leakage and clearances in the different cylinders. Having reference to the entropy diagram for adiabatic expansion (which, however, is but rarely obtained in a reciprocating engine) and for a total temperature range designated by admission and exhaust pressures, the temperature range for each of three stages containing an equal number of heat units varies so that it is greatest in the first stage, less in the second and smallest in the third. This, applied to an ex-

ample where the temperature conforms to a pressure range of 180 pounds absolute and 2 pounds absolute, we have from the entropy diagram, approximately:

Temperature drop in 1st stage = 91 degrees F.
2d stage = 82 degrees F.
3d stage = 74 degrees F.
Total = 247 degrees F.

Also from a triple-expansion engine in which the power developed in three stages was practically the same:

Temperature drop in 1st stage = 70 degrees F.
2d stage = 64 degrees F.
3d stage = 113 degrees F.

The foregoing shows a comparison of theoretical temperature drops for adiabatic expansion and those that actually occur in a triple-expansion engine for equal division of heat units in the three stages with given initial and final temperatures. Diagram, shown on Figure 4, gives an idea of pressures and power developed in a small engine.

STEAM CONSUMPTION.

Economy in fuel consumption, either coal or oil, is looked upon as one of the most important factors in the selection of class and type of machinery installations for naval vessels.

The apparently extravagant use of fuel for propulsion at reduced speeds, while largely due to more uneconomical operation of the main machinery, is to a greater extent due to the auxiliaries. It was thus pointed out in an article by Professor Hollis upon the performance of the *Charleston* that 72.2 per cent. of the coal was expended on auxiliaries and losses at 4 knots, 46.6 per cent. at 8 knots and 15.8 per cent. at 18 knots. We find, similarly, in the speed trials of the *Delaware* that 10.5 per cent. was used for the auxiliaries at 21.56 knots and 36.3 per cent. at 12.24 knots. At the same

DIAGRAM AND SYNOPSIS OF PRESSURES FROM BOILER TO CONDENSER.

FROM TESTS MADE BY CHIEF ENGINEER ISHERWOOD, U. S. N., ON A SMALL RECIPROCATING STEAM ENGINE. THEORETICAL AND HYPERBOLIC EXPANSION SHOWN IN COMPARISON.

1. Boiler pressure, . . . 163.0 pounds abs.
2. Beginning of stroke, H.P. cylinder, 141.0
3. Point of cut-off, H.P. cylinder, 125.0
4. End of stroke, H.P. cylinder, . 86.5
5. In receiver, ... 44.9
6. Beginning of stroke, L.P. cylinder, 40.0
7. Point of cut-off, L.P. cylinder, . 32.0
8. End of stroke, L.P. cylinder, . 22.0
9. Mean back pressure, L.P. cylinder, 9.2
10. Vacuum in condenser, inches mercury, 18.

Size of engine, $\frac{6 \text{ ins.} \div 10\frac{1}{2} \text{ ins.}}{8}$.

H.P. cylinder cut-off = .7 ; release = .95 . cushion = .875.
L.P. cylinder cut-off = .7 ; release = .95 ; cushion = .875.
Piston speed = 468. Cylinders unjacketed. No superheat.
Net horsepower developed by H.P. cylinder = 29.23.
Net horsepower developed by L.P. cylinder = 28.15.

Fig. 4.

time, taking corresponding speeds, the main engines used 13.38 pounds and 15.12 pounds of steam, while the auxiliaries used 42 pounds and 64 pounds of steam, respectively, all per I.H.P. hour.

DETERMINATION OF WATER RATE.

Comparison of Indicated Steam Consumption with that Actually measured on the Birmingham and Delaware.

While no accurate estimate of performance, from a thermodynamic standpoint, can be made from weight of water per I.H.P. alone, it is a highly important determination for purposes of comparison.

Lately, in all ships of the Navy, this has been done by actual weighing of the condensed steam from main and auxiliary machinery. This method is the only one known to give absolutely reliable results.

As it is not practicable, however, always to resort to these means of ascertaining steam consumption, other means, although only approximately correct, are often used, among which the steam accounted for by the indicator is of useful application. The *exact* amount of steam used by the engine is not determinable from the diagram, because that gives the *pressure of the steam* only, and does not take into account the water that has been condensed by contact with the cylinder as a result of liquefaction.

The steam consumption is naturally determined from the high-pressure cylinder, although it may also be determined from the low-pressure cylinder, provided no live steam has been admitted to this cylinder. In comparison the steam consumption calculated from the high-pressure cylinder will appear greater than that figured from the low-pressure, and the difference may be taken as a measure of the loss that occurs in transmission of the steam through the engine.

Various methods may be employed to determine steam consumption from the indicator diagrams, all of which give the "indicated steam consumption" so-called because the pressure

at cut-off and compression are obtained from the indicator card, and corresponding densities from steam tables (there being no live steam used in receivers). The method here used will assume full knowledge of the engine, such as size of cylinders and clearances, stroke, revolutions and horsepower. Its accuracy depends largely upon the ability to establish—

1. Cut-off pressure;
2. Compression pressure;
3. Quality of steam at cut-off.

The quantity of steam, Q, in pounds per hour, used by the engine is obtained from:

$$Q = 2S \times R \times \frac{a}{144} \times 60\ [d \times f + c(d - d_1)] =$$
$$.833\, R \times S \times a\ [d \times f + c(d - d_1)];$$

in which

R = Revolutions per minute;
S = Stroke in feet;
a = Area of H.P. cylinder minus one-half the area of piston rod, all in square inches;
f = Cut-off in decimal of stroke;
c = High-pressure cylinder clearance in per cent. of cylinder volume;
d = Density of steam at cut-off pressure;
d_1 = Density of steam at compression pressure.

If

I.H.P. = Total indicated horsepower developed by the engine,

$\frac{Q}{\text{I.H.P.}}$ = Pounds steam per I.H.P. hour.

Applied to the *Delaware*, we have:

1. Valve-chest pressure, pounds absolute,	268
2. Cut-off pressure, from cards, pounds abs.	225
3. Superheat, average, degrees F.,	66
4. Density, d,	.441
5. Compression pressure, from card, pounds absolute,	173

6. Density, d_1, ...3801
7. Clearance, ...162
8. Revolutions per minute, ... 128.3
9. Stroke, in feet, ... 4
10. Net area H.P. cylinder, square inches, . 1,139
11. I.H.P., ... 14,290
12. Cut-off, H.P. cylinder, ...86

$$Q = .833 \times 128.3 \times 4 \times 1{,}139\ (.86 \times .441 + .162\ (.441 - .3801)\ ) = 178{,}340 \text{ pounds per hour.}$$

$$\frac{Q}{\text{I.H.P.}} = 12.48 \text{ pounds per I.H.P. hour.}$$

No live steam used in receivers.

The constants given in the following table were derived by careful measurement and calculation from indicator cards by Mr. Stephen Rozycki, of the Bureau of Steam Engineering :

Name of ship.	Speed of ship.	Piston speed.	Constant *C*. Mean of starboard and port engines.
Delaware . .	21.56	1,027	1.072
	12.24	568	1.161
Birmingham .	24.15	1,170	1.111
	14.95	666	1.236
	10.65	492	1.364

$$C = \frac{\text{Actual water consumption}}{\text{Calculated water consumption}}.$$

It is plainly seen that the "calculated from cards" water consumption is considerably smaller than the actual, and that the difference is materially greater in engines of the *Birmingham* class than that of the *Delaware*. The reasons for the discrepancy are due, first, to water carried over with the steam unaccounted for by the card, and, secondly, to the condensa-

Table IV.

ACTUAL WATER RATE IN POUNDS PER I.H.P. PER HOUR AS MEASURED DURING TESTS OR ON TRIAL.

Name of ship.	Percentage of power.	Steam at engine.		Vacuum inches in condenser.	Cylinder ratio.	Piston speed in ft.	Water in lbs. per I.H.P. hour measured.	Superheat.	Characteristics of engine.
		Pressure by gage.	Temperature.						
			° F.					° F.	
S. S. *Iona*	86	165	373	28.2	6.71	391	13.35	...	One 3-cyl. triple-exp. engine; H.P. cyl. steam jacketed; all slide valves.
S. S. *Zenith City*	Full power..	192	384	22.85	8.2	610	15.42	...	One 3-cyl. triple exp. engine; no steam jackets; slide valves on I.P. and L.P.
H. M. S. *Hermes*	Full power..	172	377	.....	.....	.....	15.64	...	Two 4-cyl. triple-expansion engines.
S. S. *Pennsylvania*	10	229	393	.....	.....	.....	21.9	...	One cyl. quadr.-exp. eng; slide valve on L.P. cyl.; cyls. all unjacketed.
	91	237	401	22.73	12.63	530	14.4	...	
	18	.....	.....	21.16	12.63	300	20.2	...	
U.S.S. *Birmingham*	Full power..	234	395	26.5	9.6	1,150	17.4	...	Two 4-cyl. triple.-exp. engines; cyls. all unjacketed; piston valves all cyls.
	67	187	382	27.0	9.6	1,032	16.6	...	
	10	201	388	26.8	9.6	547	20.2	...	
U. S. S. *Delaware*	Full power..	285	478	26.3	7.8	1,027	13.38	61	Two 4-cyl. triple-exp. engs.; barrels of all cyls. and top and bottom of I.P. and L.P. steam jacketed.
	58.8	272	463	27.16	7.8	868	12.85	50	
	13.6	252	443	27.97	7.8	544	15.12	36	
Yacht *Idalia*	Full power..	190	383	25.5	8.0	582	18.3	0	One 4-cyl. triple-expansion engine.
	Do.	196	443	25.9	8.0	574	17.0	57	
	Do.	198	483	25.4	8.0	574	15.8	96	
	Do.	203	494	25.2	8.0	580	15.5	105	

tion due to cylinder characteristics, which is undoubtedly greater in the *Birmingham's* engines than in the *Delaware's*.

In Figure 5 are shown curves of relative water rates per I.H.P. of U. S. S. *Delaware* and *Birmingham*.

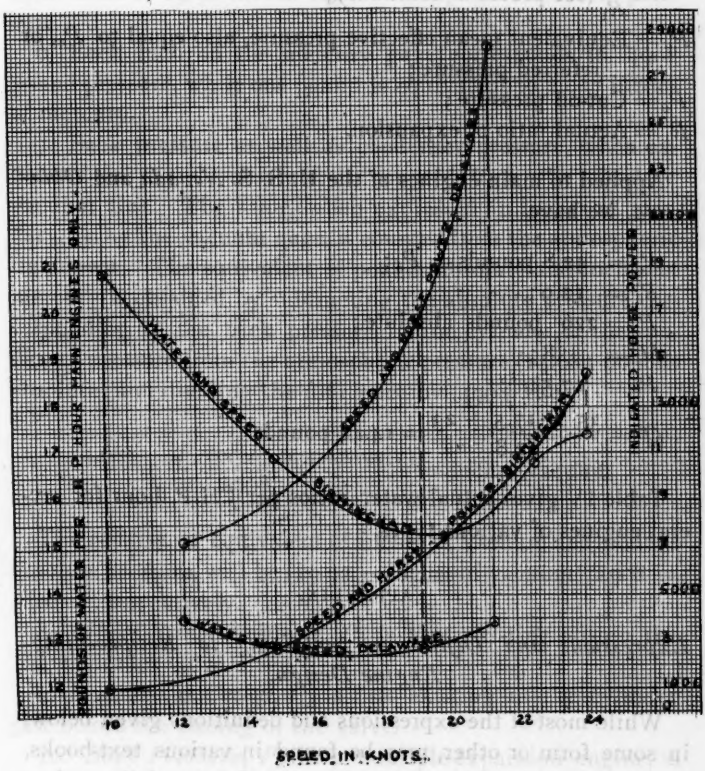


Fig. 5.

The following semi-empirical formula for approximate figuring of steam consumption may be applied with a fair degree of accuracy:

$$Q^1 = \frac{34\,P_r + 23}{P_Q};$$

in which

Q^1 = Steam consumption per I.H.P. hour;
P_T = Absolute terminal pressure from ideal indicator diagram
$= \frac{P_o}{R_1}$ (see previous notations);
P_Q = Equivalent mean effective pressure, also equal to P_R, or referred pressure;
P_o = Cut-off pressure;
R_1 = Actual ratio of expansion.

Applied to main engines of the U. S. S. *Nevada* and *Oklahoma*, we have

P_Q = 42.8 pounds = P_R;
R_1 = 12.9;
P_o = 226 pounds absolute;
H_c = .8;
P_T = 17.5;

$$Q^1 = \frac{35 \times 17.5 + 23}{42.8} = 14.44 \text{ pounds.}$$

Table IV gives actual water ratios per I.H.P. hour for several engines of various types.

APPENDIX.

Expressions and Definitions pertaining to Reciprocating-Engine Design.

While most of the expressions and definitions given below, in some form or other, may be found in various text-books, their enumeration here will be found useful especially as they are worded directly to apply to the subject discussed.

1. *Pressure by Gage* is that above the atmospheric line.

2. *Absolute Pressure* is that above the line of no pressure, and at sea level is equal to gage pressure plus 14.684 pounds per square inch.

3. Any pressure below the *atmospheric* indicates vacuum.

4. *Vacuum* is measured in either pounds per square inch, absolute pressure, or in inches of mercury, counted from the line of absolute vacuum.

5. *Absolute Vacuum*, state of no pressure at sea level, is equivalent to 29.92 inches Hg. or zero pounds absolute pressure.

6. Each inch on the mercury gage corresponds to .4908 pound pressure, while one pound is equivalent to 2.037 inches of mercury.

7. Back pressure as well as condenser pressure, when expressed in pounds absolute, is counted from the absolute vacuum line. The same, when expressed in inches of mercury, or by the vacuum gage, is counted from the atmospheric line down. The equivalent of pounds absolute and inches of mercury is obtained by subtracting the absolute pressure corresponding to the back pressure in the cylinder or the condenser pressure from the atmospheric pressure (14.684 pounds absolute) and dividing the difference by .4908, thus:

$$\frac{14.684 - P_b \text{ (or } P_c \text{ respectively)}}{.4908} = \text{inches vacuum.}$$

The reverse, or equivalent of inches of vacuum and pounds absolute pressure, will be (29.92 — inches of vacuum) .4908 = P_b or P_c respectively.

8. *Initial Steam Pressure*, P_1, is that existing within the H.P. cylinder valve chest.

9. *Pressure at Cut-off*, P_c, is that existing in the cylinder just before the closing of the valve.

10. *Back Pressure*, P_b, in a multiple-expansion engine, is the pressure of the exhaust in the low-pressure cylinder.

11. *Terminal Pressure*, P_t, is the theoretical expression for the pressure existing at the end of the stroke in the low-pressure cylinder. It is measured by the distance between the absolute vacuum line and the assumed extension of the expansion curve at the end of the stroke. It is equivalent to the cut-off pressure divided by the actual ratio of expansion.

12. *Condenser Pressure*, P_c, is that existing in the condenser.

13. *The Theoretical Mean Pressure*, P_M, in a multiple-expansion engine, is obtained from the formula for hyperbolic expansion,

$$P_M = P_1 \frac{1 + \text{hyp. log. } R_1}{R_1},$$

where

P_1 = initial pressure in pounds absolute;
R_1 = actual ratio of expansion.

14. *The Theoretical Mean Effective Pressure*, P_E, is obtained by subtracting from P_M the back pressure P_B,

$$P_E = P_M - P_B.$$

15. *The Referred Pressure*, P_R, *Equivalent or Actual Mean Effective Pressure*, P_Q, is obtained either by the addition of each of the individual mean effective pressures with respect to each cylinder, or by considering the whole power of the engine were developed in the low-pressure cylinder when acted upon by this pressure. The expressions will be, for triple-expansion engines, respectively:

$$P_R = \frac{\text{I.H.P.} \times 33{,}000}{A \times S};$$

$$P_Q = P_L + \frac{P_1}{\text{Ratio L.P. to I.P.}} + \frac{P_H}{\text{Ratio L.P. to H.P.}}$$

where

$P_R = P_Q$;
I.H.P. = Total indicated horsepower developed by the engine;
A = Area of L.P. cylinder in square inches with allowance made for area of piston rod;
S = Piston speed in feet per minute;
P_L = Mean effective pressure, from cards, in L.P. cylinder;
P_1 = Mean effective pressure, from cards, in I.P. cylinder;
P_H = Mean effective pressure, from cards, in H.P. cylinder.

16. *Cylinder Ratio*, r_n, in engines without tail rods

$$r_n = \frac{2\,A_L - d}{2\,A_H - d},$$

where

A_L = Area of L.P. cylinder;
A_H = Area of H.P. cylinder;
d = Area of piston rod.

17. *Nominal Ratio of Expansion*, R,

$$R = \frac{\text{L.P. cylinder capacity}}{\text{H.P. cylinder capacity at cut-off}} = \frac{2\,A_L - d}{(2\,A_H - d)\,H_c}.$$

18. *Actual Ratio of Expansion*, R_1,

$$R_1 = \frac{(2\,A_L - d)(1 + c)}{(2\,A_H - d)\,(H_c + c_1)},$$

where

A_L, A_H and d are as before, and

c = L.P. cylinder volumetric clearance;
c_1 = H.P. cylinder volumetric clearance;
H_c = H.P. cylinder cut-off.

19. *Adiabatic* expansion or compression is that in which the steam, in expanding or when being compressed, neither receives nor rejects heat. Such expansion does not occur in the ordinary engine. The equation for adiabatic expansion is PV^γ = constant, where γ is the ratio between the two specific heats, P = pressure and V = volume.

20. *Hyperbolic* expansion (isothermal), or compression is that in which the temperature of the steam during expansion or compression is kept constant. The curve of hyperbolic expansion is a rectangular hyperbola, the asymptotes of which are the lines of pressure and volume. The equation for hyperbolic expansion is PV = constant.

21. *Mariotte's law* states that the volume of a given mass of gas varies inversely as the pressure, provided the tempera-

ture be kept constant, or $PV =$ constant. This is also the equation for hyperbolic expansion.

22. *For wet steam* $PV^{\frac{15}{16}} =$ constant.

23. *For dry saturated* and *superheated steam* above 120 pounds pressure $PV =$ constant. The latter equation is considered to represent the condition of expansion taking place in marine engines generally, and is used in the determination of mean pressures.

24. *Piston speed*, S, is the number of feet traveled by the piston per minute.

$$S = L \times N;$$

where

$L =$ length of stroke in feet;
$N =$ number of single strokes per minute;
$= 2$ times number of revolutions.

25. *Indicated Horsepower*, I.H.P., is the measure for the work done by the steam in the cylinder and is the sum of the work done in all the cylinders in multiple-cylinder engines.

$$\text{I.H.P.} = \frac{A \times P \times S}{33{,}000} = \frac{P \times L \times A \times N}{33{,}000}.$$

S, L and N denote quantities as previously given, while $A =$ net area of any cylinder, and $P =$ actual mean effective pressure.

26. *Brake Horsepower*, B.H.P., is the indicated horsepower minus the friction of the engine,

B.H.P. $= .9$ to $.93$ I.H.P. (more or less).

27. *Mechanical Efficiency*, E_M, is the ratio of brake and indicated horsepower.

$$E_M = \frac{\text{B.H.P.}}{\text{I.H.P.}}.$$

28. *Thermodynamic Efficiency* E_T is the ratio between the heat converted into work and the heat received by the engine.

$$E_T = \frac{T_1 - T_2}{T_1}$$

where

T_1 = absolute temperature of the entering steam;
T_2 = absolute temperature of the exhaust steam.

This efficiency in marine engines does not exceed 17 per cent.

29. *Cylinder Constants* are represented by the decimal fraction of the net cylinder area and 33,000, when the indicated horsepower is given by formula

$$\text{I.H.P.} = \frac{A \times P \times S}{33{,}000}.$$

30. *Card Effect*, F, is the ratio of *actual* referred pressure, P_R, and *theoretical* mean effective pressure, P_E.

31. The mechanical equivalent of 1 B.T.U. is 778 foot-pounds.

1 foot-pound	=	$\frac{1}{778}$	B.T.U.;
1 horsepower	=	1,980,000	foot-pounds per hour;
	=	33,000	foot-pounds per minute;
	=	550	foot-pounds per second;
	=	2,545	B.T.U. per hour;
	=	42.41	B.T.U. per minute;
	=	0.77	B.T.U. per second.

SCREW PROPELLER DESIGN.

The Influence of the Ratio Between Greatest Immersed Beam and Length on Load Water Line on Block Coefficient and on the Choice of Slip Chart to Use for any Given Vessel.

By Captain C. W. Dyson, U. S. N., Member.

After the publication of the article on "Screw Propeller Design," prepared by the writer and published in the August, 1911, number of the Journal of the American Society of Naval Engineers, the "London Engineer," in its review, called attention to the fact that it is not the Block Coefficient alone which should be used in determining the proper propeller for a ship.

This point had been thoroughly understood by the writer, but at the time no practical way of graphically illustrating corrections to be made for variations in relative dimensions of any particular vessel from those of the vessels which were used in developing the Charts of Design given in the article referred to was in sight.

By adopting the system of "Parent Lines," as advocated by Naval Constructor Taylor in his late book on "Speed and Power of Ships," a Guide Chart for Correction of Block has been produced and is herewith submitted as the most practicable method of graphically illustrating the correction of block to be used in the selection of the slip charts where any ordinary departure from the relative dimensions of the standard basic vessels occurs.

The action of a propeller working in the wake of a vessel is

dependent upon the motion of the water in which it operates. This depends on the form of the afterbody, and for any constant form of afterbody, the motion of the water at the propeller will be practically constant, no matter what the ratio of length to beam of the vessel, within the practical limits of ship construction.

DEVELOPMENT OF THE CORRECTIVE CHART.

If from the vessel of "Parent Lines" we take away the parallel-sided middle body and replace it with one of greater or less length than the original, at the same time retaining the beam and immersed midship section of the parent ship, we produce the following results:

1. Modification of the effective horsepower required for any given speed due to change in residual resistance and skin friction.

2. Increase or decrease in the displacement, according to whether the middlebody has been lengthened or shortened.

3. Increase or decrease of the block coefficient of fineness and the longitudinal (prismatic) coefficient according to whether the middlebody has been lengthened or shortened.

These changes are indicated by the following formulas:

Let D = Original displacement;
d = Displacement added or taken away;
A = Area of immersed midship section;
B = Greatest immersed beam of parent ship;
H = Draught of parent ship;
L = Original length of ship on L.W.L.;
l = Length added to or taken away from middle body;
$B.\ C.$ = Block coefficient of fineness;
$L.\ C.$ = Longitudinal (prismatic) coefficient;
$I.\ C.$ = Coefficient of fullness of midship section;

all dimensions being in feet, square feet and tons.

Amount of displacement added or subtracted by altering the length of the middle body equals—

$$d = \frac{l \times A}{35};$$

and the new displacement is

$$D \pm d.$$

The original block coefficient was

$$\frac{D \times 35}{L \times B \times H} = B.\ C._1$$

The new block coefficient is

$$\frac{(D \pm d)\,35}{(L \pm l) \times B \times H} = B.\ C._2$$

The original longitudinal coefficient was

$$\frac{D \times 35}{L \times A_1} = L.\ C._1 = \frac{D \times 35}{L \times B \times H \times I.\ C._1}$$

while the new longitudinal coefficient is—

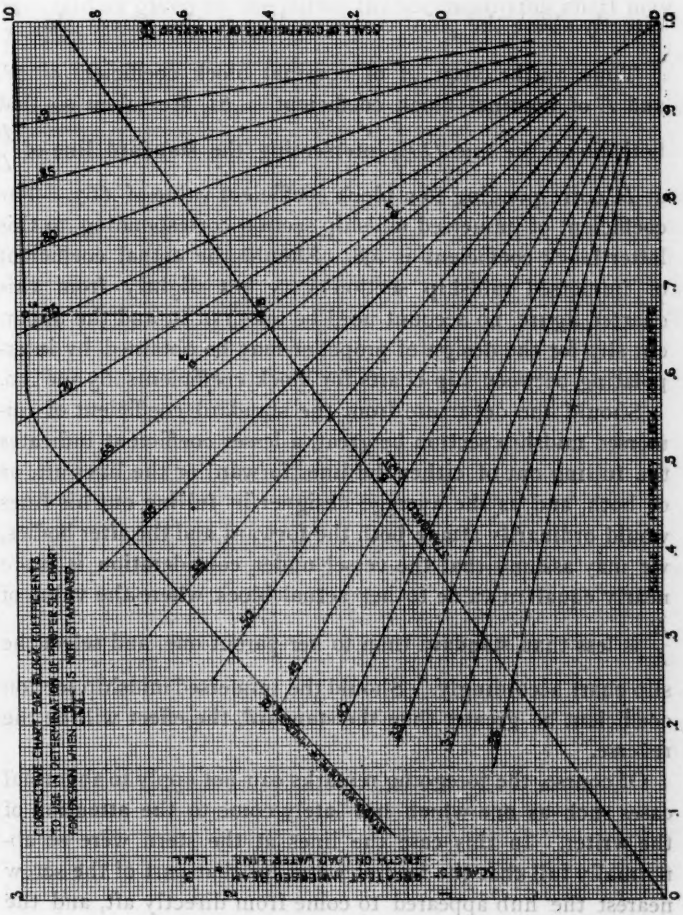
$$\frac{(D \pm d)\,35}{(L \pm l) \times A_1} = L.\ C._2.$$

In the course of these changes, the conditions of the fore and afterbodies of the vessel have remained unchanged, and as the motion of the water under the stern depends upon the lines of the afterbody, the propeller of the parent ship and the propellers of these derived vessels are working under identically the same conditions of inflowing water and should therefore be designed for the same conditions of thrust and slip.

If the block coefficients for a series of such desired vessels be plotted on a chart having for abscissae values, block coefficients from 0 to 1.0, and for ordinates values of $\frac{B}{L}$, these points will be found to plot in a straight line passing through the axis of the abscissae at $B.C. = 1.0$, and through the point on

the chart corresponding to the *B.C.* and the value $\frac{B}{L}$ of the parent ship.

This fact having been established by trial, all that is necessary in constructing the corrective chart for block coefficients is to establish the line of parent (standard) $\frac{B}{L}$ and of


CORRECTIVE CHART FOR BLOCK COEFFICIENTS.
TO USE IN DETERMINATION OF HORSE POWER BY CHART
FOR WHICH WHEN $\frac{B}{L}$ IS NOT STANDARD
SCALE OF CENTRE BNTS OF IMMERSED
SCALE OF IMMERSED BLOCK COEFFICIENTS
TEST
CURVES
STANDARD $\frac{B}{L}$ PARENT SHIP
SCALE OF IMMERSED BLOCK COEFFICIENTS
LENGTH ON LOAD WATER LINE = L

parent (standard) coefficient of immersed midship section. Then all vessels of equal and approximately equal fineness of fore and afterbodies may be considered to lie on a line drawn through the parent-ship point and the point marked D on the accompanying chart, while the immersed midship section check on variation of fulness of these bodies will be on the curve of standard coefficient of immersed midship section directly above the point corresponding to the parent ship $\frac{B}{L}$.

Thus, on the chart, A of nominal block coefficient = .78, and E of nominal block coefficient = .61, are taken so that both lie on the line $D\,E$, intersecting the standard line of $\frac{B}{L}$ at B, corresponding to a block coefficient value of .667. The coefficient of immersed midship section corresponding to this latter block coefficient is .98. Should the actual coefficient of immersed midship section vary but slightly from this charted figure, no account need be taken of it, and the apparent slip for the design of propeller may be obtained by interpolating between slip charts for block coefficients .65 and .70.

Should the departure from the standard coefficient of immersed midship section be great, a lesser coefficient indicates the fulling out of either the lines forward or the lines aft, or of both, and as the average designer in fulling out his lines would ordinarily attack both the forward and the after bodies, we may assume that the vessel under consideration is more nearly equal in value to her actual block where the value of $\frac{B}{L}$ is less than standard than to her parent one, and select the slip chart accordingly. Should the immersed midship section coefficient be greater than the standard, the effect will be the reverse.

Of course, the foregoing remarks can not apply to abnormal cases such as one which has lately come to the attention of the writer. In this case the lines at the stern were so abnormally full that the feed of water for that part of the screw nearest the hub appeared to come from directly aft, and the

water leaving the screw was thrown off radially. The result was that while the model towed easily, it was impossible to fit any propeller that would give the required speed, although the power developed by the propelling engines was very high, as compared with the necessary effective horsepower.

If, however, the ship is normal in her lines, with no wide departure in coefficient of immersed midship section from the standard as given by the curve, the accompanying chart may be used with confidence.

REPORT OF TEST OF A BOILER FEED PUMP OF BOSTON NAVY YARD DESIGN, AT THE ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

The object of the test was to determine the efficiency, water rate per I.H.P., slip, and the duty of this pump at different speeds, discharge pressures and temperatures of water handled.

The pump, shown on Plates I and II, is of the vertical, simplex, solid-piston type, and is of the following dimensions:

Number of steam cylinders	1
Diameter of steam cylinder, inches	16
piston rod, inches	3
Area of top end of steam piston, square inches	201.06
bottom end of steam piston, square inches	193.99
Nominal stroke, inches	18
Number of water plungers	1
Diameter of water plunger, inches	10
rod, inches	3
Area of top end of water plunger, square inches	71.47
bottom end of water plunger, square inches	78.54
Nominal stroke, inches	18
Water-plunger packing	solid white metal.
Number of suction valves	8
Area of suction valves (clear), each, square inches	6.96
Maximum lift, inches	$\frac{9}{16}$
Type	metal composition, solid.
Number of discharge valves	8
Area of discharge valves (clear), each, square inches	6.96
Maximum lift, inches	$\frac{9}{16}$
Type	metal composition, solid.
Area of steam pipe, square inches	3.14
exhaust pipe, square inches	4.59
suction pipe, square inches	19.63
discharge pipe, square inches	28.27

The main valve is of the floating type, sliding over a sleeve, through which the steam is admitted, and operated by a pilot valve in front of the steam cylinder; this pilot valve being operated by levers and tappets from the piston rod.

On starting the tests it was found that the pump would work satisfactorily for a short time and then stop, and could not be made to move without removing the main steam valve and then replacing it, when the pump would function for a short time again. On examining the main steam valve it was found that this valve was of such a length as to allow overtravel of one-half inch in each direction, the valve probably being shortened to allow for a steam cushion at the end of travel to prevent striking the valve-chest heads. As long as the cushion effect held good the pump operated, but at times the momentum of the shifting valves would overcome this cushion and strike the valve-chest head, and the pump would stop.

To shorten the stroke of this valve three-eighth inch rings were attached by countersunk screws to each end of the main valve. On reassembling it was found that while the pump would operate, the valve would strike at each stroke. This difficulty was remedied by inserting taper wedges in the steam grooves at each end of the steam sleeve, so that as the main valve neared the end of the stroke the steam passage became more and more restricted, thus forming the necessary cushion.

After running for two days the main valve was removed for inspection, and on reassembling the pump refused to operate, and, as no trouble with the mechanism could be discovered, it was suggested that possibly the steam sleeve was out of center, thus binding the main valve against the steam-chest walls. The steam-pipe flange was removed and the steam sleeve measured; it was found to be .004 inch out of center. This could be corrected by setting off on certain of the steam-chest cover bolts, and tightening up on others, due to packing in the joint allowing one bolt to be squeezed down tighter than the one diametrically opposite. The steam sleeve was also found to be slightly warped. The sleeve was re-

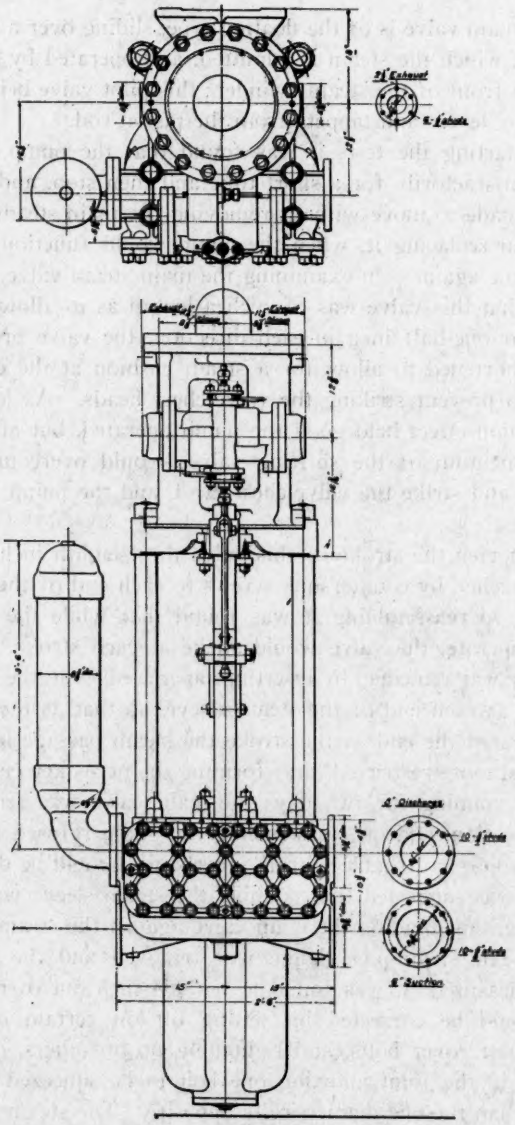


Plate I.

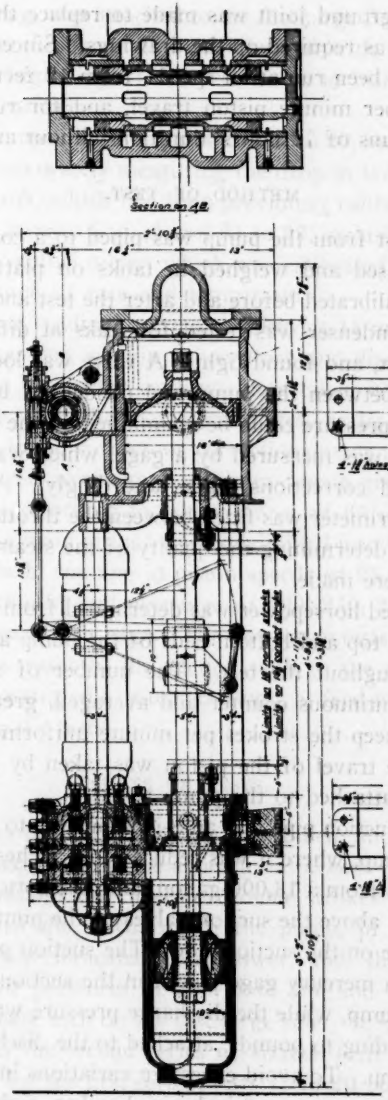


Plate II.

ground, and ground joint was made to replace the 1/32 inch packing joint as required on the drawings. Since the changes the pump has been run at all speeds, from 12 feet per minute to 100 feet per minute piston travel, and for runs of short duration to runs of 72 hours' duration without any difficulty.

METHOD OF TEST.

The exhaust from the pump was piped to a condenser, the steam condensed and weighed in tanks on platform scales, which were calibrated before and after the test and found correct. The condenser was tested for leaks at different times during the test, and found tight. A valve was located on the exhaust line between the pump and condenser, by means of which a back pressure could be maintained on the pump. The back pressure was measured by a gage, whicn was calibrated each day, and corrections made accordingly. A Carpenter throttling calorimeter was fitted between the throttle valve and the pump, for determining the quality of the steam, for which corrections were made.

The indicated horsepower was determined from cards taken from both the top and bottom ends of the pump at 15-minute intervals throughout the test. The number of strokes was taken by a continuous counter and averaged, great pains being taken to keep the strokes per minute uniform throughout the test. The travel of the piston was taken by a recording stroke meter attached to the pump rod.

The pump suction pipe was an 8-inch pipe up to within four feet of the pump, where it was reduced to 6 inches. The suction was taken from a 13,000-gallon tank, the bottom of which was three feet above the suction valves of the pump, thus putting a pressure on the suction side. The suction pressure was measured by a mercury gage placed in the suction pipe at the entry to the pump, while the discharge pressure was measured by a gage, reading to pounds, attached to the discharge chamber of the pump. To avoid excessive variations in the suction and discharge pressures a 6-inch air chamber, eight feet long, was attached to the suction pipe, while an 8-inch air chamber,

eighteen feet long, was attached to the discharge line. The air chamber on the discharge line was charged with compressed air before the beginning of the tests so as to insure the full benefit of the chamber.

The water pumped in tests 1 to 4 and 17 to 20, inclusive, was determined by accurately measuring the drop in level of water in the supply tank, which had been previously calibrated. The water pumped in tests 5 to 16 and 21 to 32, inclusive, was determined by a venturi meter, which was calibrated before and after each change of rate and temperature.

The following readings were also taken at 15-minute intervals: Boiler pressure, suction temperature and discharge temperature, while barometer readings were taken before beginning each test. All thermometers, gages, scales, etc., were calibrated and found correct, or corrections made accordingly.

Thirty-two tests were run in all, in two main groups of sixteen tests each according to temperatures of 65 and 150 degrees F. These main groups were subdivided into four groups of four tests each, running at piston speeds of 25, 50, 75 and 100 feet per minute, and pressures of 200, 250, 300 and 325 pounds per square inch gage. The hot-water tests were made by heating the water in the suction tank to the temperature required by passing the water through a feed-water heater before starting the tests.

RESULTS OF TESTS.

Table I gives the average of the data taken, and the results as calculated according to the appendix attached.

Plate III contains the curves of double strokes plotted successively against the discharge of the pump, in pounds per hour, steam per indicated horsepower, and the indicated horsepower, for pressures of 200, 250, 300 and 325 pounds (gage) per square inch on the discharge side. It will be noted that the slip in the second series of sixteen tests is much in excess of that for the first series; tests 17 to 20 (the last four tests to be run) being about double that for tests 1 to 4 (the first four tests to be run). This increase in slip was due to

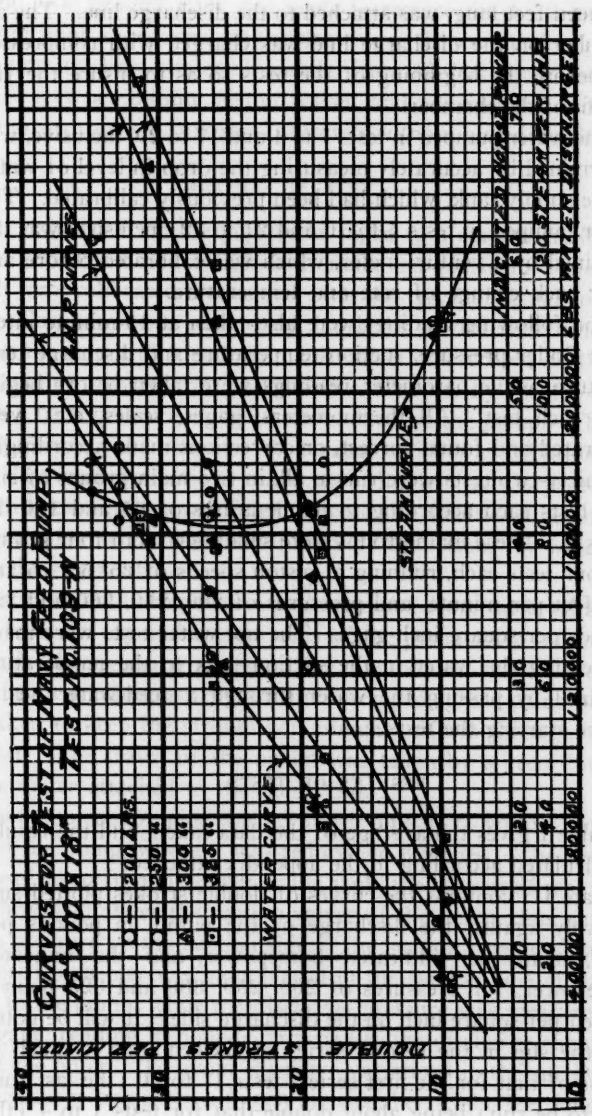

Curves for Test of Worthington Duplex Steam Pump
Water Curve
Steam Curve
Boiler Pressure
Plate III.

TABLE I.—TEST OF NAVY-TYPE B

				Data from steam end.																		
Test number.	Date, October, 1911.	Duration of test, in hours.	Barometer (corrected).	Steam pressure at boiler, pounds gage.	Steam pressure in steam chest, lbs. gage.	Steam pressure in steam chest, lbs. absolute.	Back pressure, pounds absolute.	M.E.P. of steam cylinder, average two ends.	Number double strokes per minute.	Length of stroke, in feet.	I.H.P. developed by steam cylinder.	Steam consumed per hour, in pounds.	Quality of steam, per cent.	Dry steam consumed per hour, in pounds.	Dry steam consumed per minute, in pounds.	Total B.T.U.s supplied pump per hour.	Total B.T.U.s rejected by pump, and used in heating feed water, per hour.	Number of B.T.U.s consumed by pump, per hour.	Pressure in main on delivery side, pounds gage.	Press. on suction side, ins. mercury.	Pressure corresponding to that given in column 21, pounds gage.	Vert. distance between the two gages.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22	23
1	17	1	30.03	276.4	93.8	108.55	14.78	78.5	10.13	1.268	12.52	1390	98.6	1370.5	22.84	1627941	1239519	388422	200	8	3.93	0
2	17	1	30.00	275.6	112.6	127.34	14.94	97.5	9.66	1.25	14.10	1593	98.5	1569.1	26.15	1868135	1419463	448672	250	8	3.93	0
3	17	1	30.07	275.4	133.6	148.37	15.54	115.9	10.11	1.26	17.72	1953	98.4	1921.8	32.03	2293109	1740185	552924	300	8	3.93	0
4	17	1	30.65	279.2	140.6	155.66	15.96	124.1	9.95	1.238	18.34	2027	98.8	2002.7	33.38	2391340	1814638	576702	325	8	3.93	0
5	18	1	29.75	274.0	96.8	111.42	14.63	81.7	18.40	1.353	24.38	2218	99.1	2198.0	36.63	2611887	1987379	624508	200	8	3.93	0
6	18	1	29.82	274.6	123.8	138.45	14.67	100.0	19.31	1.324	30.74	2613	99 0	2586.9	43.11	3083678	2339189	744489	250	8	3.93	0
7	18	1	29.78	271.8	147.4	162.03	15.34	119.7	19.43	1.330	37.02	3121	98.7	3080.4	51.34	3680199	2788480	891719	300	8	3.93	0
8	18	1	29.78	272.6	160.2	174.83	15.81	128.5	18.57	1.340	38.34	3171	98.5	3123.4	52.06	3735259	2830216	905043	325	8	3.93	0
9	18	1	29.78	269.6	124.4	139.03	15.17	81.6	26.58	1.397	36.34	3166	99.0	3134.3	52.24	3736447	2836471	899976	200	8	3.93	0
10	18	1	29.80	254.0	149.0	163.64	15.61	101.1	26.90	1.400	45.22	3793	98.5	3736.1	62.27	4464128	3383448	1080680	250	8	3.93	0
11	19	1	29.89	275.8	174.0	188.69	15.99	125.2	25.96	1.408	54.74	4409	98.5	4342,9	72.38	5199154	3935413	1263741	300	8	3.93	0
12	19	1	29.90	277.6	183.2	197.89	16.20	133.3	25.98	1.418	58.70	4630	98.8	4574.4	76.24	5479676	4146395	1333281	325	8	3.93	0
13	19	1	29 90	274.6	128.0	142.69	15.22	81.7	33 10	1.420	46.30	3864	98.7	3813.8	63.56	4548188	3451621	1096567	200	8	3.93	0
14	19	1	29.89	269.0	176.6	191.26	16.06	102.7	34.76	1.420	60.70	5291	98.8	5227.5	87.13	6259055	4737498	1521557	250	8	3.93	0
15	19	1	29.90	257.4	198.6	213.29	16.47	123.7	30.45	1.465	65.96	5553	98.7	5480.8	91.35	6572039	4969794	1602245	300	8	3.93	0
16	20	1	30.09	270.2	211.4	226.18	17.20	133.5	31.43	1.434	72.18	6097	98.7	6017.7	102.96	7221288	5462152	1759136	325	8	3.93	0
17	20	1	30.07	270.0	92.4	107.17	15.50	75.4	8.18	1.368	10.14	1295	99.2	1284.6	21.41	1525549	1163094	362455	200	8	3.93	0
18	20	1	30.07	273.0	112.0	126.77	15.30	95.3	8.73	1.368	13.68	1577	99.3	1566.0	26.10	1864244	1417476	446768	250	8	3.93	0
19	21	1	30.07	276.2	137.2	151.97	15.09	114.4	8.66	1.338	15.94	1778	98.2	1746.0	29.10	2084026	1579870	504156	300	8	3.93	0
20	21	1	30.05	273.4	148.0	162.76	15.84	123.8	9.08	1.327	17.88	1866	98.6	1839.9	30.66	2198252	1666845	531407	325	8	3.93	0
21	23	1	30.06	275.4	105.8	120.56	15.22	77.3	19.08	1.350	24.02	2339	99.0	2315.6	38.59	2754789	2095670	659119	200	8	3.93	0
22	23	1	30.05	271.0	128.0	142.76	15.47	95.7	18.91	1.360	29.54	2649	98.7	2614.6	43.58	3118072	2367219	750853	250	8	3.93	0
23	23	1	30.03	273.4	152.2	166.96	15.88	116.5	18.53	1.370	35.50	3012	98.9	2978.9	49.65	3560345	2698855	861490	300	8	3.93	0
24	23	1	30.04	275.0	165.0	179.76	16.13	123.3	18.05	1.360	36.32	3181	98.3	3126.9	52.12	3740985	2834012	906973	325	8	3.93	0
25	24	1	30.35	275.4	114.8	129.71	15.48	79.4	25.98	1.395	34.62	3039	98.2	2984.3	49.74	3554122	2702069	852053	200	8	3.93	0
26	24	1	30.34	275.2	142.8	157.70	15.91	98.1	25.50	1.400	42.76	3571	98.1	3503.2	58.39	4183707	3174056	1009651	250	8	3.93	0
27	24	1	30.33	274.2	163.8	178.70	16.18	115.7	25.93	1.410	50.98	4012	98.4	3947.8	65.80	4722647	3578262	1144385	300	8	3.93	0
28	24	1	30.33	275.4	185.6	200.50	17.01	129.2	26.61	1.414	58.28	4277	98.1	4195.7	69.93	5027126	3807291	1219835	325	8	3.93	0
29	25	1	30.46	275.8	123.6	138.56	15.65	80.7	35.68	1.442	49.76	3864	98.8	3817.6	63.63	4550844	3457411	1093433	200	8	3.93	0
30	25	1	30.45	277.0	155.2	170.16	16.18	100.7	33.93	1.436	58.84	4551	99.0	4505.5	75.09	5385953	4083781	1302172	250	8	3.93	0
31	25	1	30.40	276.2	199 6	214.53	17.29	122.1	33.55	1.430	70.34	5619	98.3	5523.5	92.06	6623481	5014251	1609230	300	8	3.93	0
32	25	1	30.38	276.8	208.0	222.92	17.57	130.7	32.25	1.450	73.40	5489	98.7	5417.6	90.29	6500038	4919926	1580112	325	8	3.93	0

Data from pump end.										Results.								
Press. on suction side, ins. mercury.	Pressure corresponding to that given in column 21, pounds gage.	Vert. distance between the two gages.	Total pressure of pump, in pounds.	Number double strokes per minute.	Total feet of piston travel, per hour.	Water delivered per hour, in pounds.	Theoretical pump discharge from plunger displacement, in pounds per hour.	Slip, calculated, pounds per hour.	Temperature of suction water, degrees F.	Duty, calculated from column 19, foot-pounds.	Percentage of slip.	Theoretical capacity, in gallons per hour.	Actual capacity, in gallons per hour.	Efficiency, per cent.	Pounds dry steam, per I.H.P. hour.	Pounds dry steam per 1,000 pounds of water pumped.	Total B.T.U.s consumed by pump per hour, assuming feed to be heated to 200 degrees F.	Duty, calculated from column 38, foot-pounds.
21	22	23	24	25	26	27	28	29	30	31	32	33	34	35	36	37	38	39
8	3.93	0	196.07	10.13	1542	38388	50495	12107	68.9	447001 17	23.98	5998.8	4560.5	70.03	109.5	35.7	1400272	12309383
8	3.93	0	246.07	9.66	1450	34074	47501	13427	67.0	43108161	28.27	5643.1	4048.0	69.28	111.3	46.1	1608181	12026895
8	3.93	0	296.07	10.11	1530	34445	50128	15683	66.0	42546357	31.29	5955.2	4092.1	67.05	108.5	55.8	1975167	11910335
8	3.93	0	321.07	9.95	1478	32968	48425	15457	65.9	42339752	31.92	5752.9	3916.6	67.24	109.2	60.8	2058844	11859771
8	3.93	0	196.07	18.40	2988	83515	97903	14388	65.5	60484411	14.70	11630.9	9921.6	78.25	90.2	26.3	2245148	16824280
8	3.93	0	246.07	19.31	3070	84480	100605	16125	65.0	64411135	16.03	11951.9	10036.2	78.79	84.2	30.6	2653005	18075119
8	3.93	0	296.07	19.43	3102	83032	101658	18626	64.7	63594433	18.32	12077.0	9864.2	77.37	83.2	37.1	3169532	17891715
8	3.93	0	321.07	18.57	2984	76950	97792	20842	64.6	62971867	21.31	11617.7	9141.7	75.08	81.5	40.6	3218882	17705603
8	3.93	0	196.07	26.58	4456	123726	146035	22309	64.5	62179472	15.28	17349.0	14698.6	77.77	86.2	25.3	3214820	17406894
8	3.93	0	246.07	26.90	4520	120819	148132	27313	64.5	63460496	18.44	17598.0	14353.3	76.60	82.6	30.9	3846102	17831167
8	3.93	0	296.07	25.96	4388	121475	143797	22322	65.0	65649354	15.52	17083.1	14431.2	76.55	79.3	35.8	4481572	18512205
8	3.93	0	321.07	25.98	4422	116504	144914	28410	64.8	64718184	19.61	17215.8	13840.7	74.24	77.9	39.3	4721535	18275311
8	3.93	0	196.07	33.10	5640	173012	184838	11826	64.5	71360525	6.40	21958.8	20553.8	85.36	82.4	22.0	3915430	19985441
8	3.93	0	246.07	34.76	5920	180133	194042	13909	64.0	67200173	7.17	23052.2	21399 8	85.07	86.1	29.0	5392428	18961568
8	3.93	0	296.07	30.45	5354	158468	175465	16997	64.5	67548276	9.69	20845.2	18826.0	82.87	83.1	34.6	5665171	19104258
8	3.93	0	321 07	31.43	5408	160676	177223	16547	65.0	67648591	9.34	21054.1	19088.3	83.27	83.4	37.5	6227110	19110481
8	3.93	0	196.07	8.18	1344	22412	42519	20107	154.8	27966845	47.29	5051.3	2662.5	50.49	126.7	57.3	1310152	7737059
8	3.93	0	246.07	8.73	1434	21760	45445	23685	152.0	27646628	52.12	5398.9	2585.1	45.60	115.1	72.0	1602910	7705753
8	3.93	0	296.07	8.66	1392	20950	44230	23280	147.4	28380544	52.63	5254.5	2488.9	45.33	109.5	83.3	1796082	7966352
8	3.93	0	321.07	9.08	1446	19961	46025	26064	143.9	27820324	56.63	5467.8	2371.4	41.76	102.9	92.2	1893561	7807467
8	3.93	0	196.07	19.08	3098	77668	98178	20510	152.0	53296081	20.89	11663.5	9227.0	73.86	96.4	29.8	2369033	14228185
8	3.93	0	246.07	18.91	3094	78434	98193	19759	149.7	59294550	20.12	11665.3	9318.0	76.12	88.5	33.3	2684285	16585977
8	3.93	0	296.07	18.53	3044	74879	96362	21483	153.8	59362396	22.29	11447.8	8895.6	72.76	83.8	39.8	3065252	16683819
8	3.93	0	321.07	18.05	2954	69039	93795	24756	148.8	56377692	26.39	11142.8	8201.8	71.10	86.1	45.3	3225311	15853678
8	3.93	0	196.07	25.98	4352	113630	137782	24152	153.5	60317507	17.53	16368.5	13499.2	74.98	86.2	26.3	3060820	16790831
8	3.93	0	246.07	25.50	4294	111759	136362	24603	148.3	62831375	18.04	16199.8	13277.0	74.93	81.9	31.4	3606353	17590557
8	3.93	0	296.07	25.93	4400	112328	139284	26956	153.9	67037451	19.35	16546.9	13344.6	76.00	77.4	35.2	4071057	18844406
8	3.93	0	321.07	26.61	4518	113025	143755	30730	144.7	68624720	21.38	17078.1	13427.4	72.54	72.0	37.1	4337220	19300574
8	3.93	0	196.07	35.68	6170	173799	195423	21624	153.1	71890595	11.07	23216.3	20647.3	79.78	76.7	22.0	3916550	20070610
8	3.93	0	246.07	33.93	5846	166022	185849	19827	146.6	72370676	10.67	22078.9	19723.4	80.89	76.6	27.1	4636877	20323823
8	3.93	0	296.07	33.55	5764	159063	182569	23506	153.0	67507601	12.88	21689.2	18896.7	78.00	78.5	34.7	5714298	19011129
8	3.93	0	321.07	32.25	5610	150215	178180	27965	148.1	70409716	15.69	21167.8	17845.5	76.55	73.8	36.1	5603391	19854983

the wear of the white-metal plunger, which, on measuring before and after the test, showed a wear of 1/16 inch after the pump had completed 171,756 feet of piston travel, when working under pressures of 200, 250, 300 and 325 pounds.

On examining the monel-metal plunger rod and pump-cylinder liner, they were found to be in as good condition as when the test was started, although the pump had been required to handle salt water at a temperature of 150 degrees F.

The pump handled water at 150 degrees F. equally as well as at 65 degrees F., but at all temperatures of water when running above 80 feet piston speed there was a tendency for the pump to hammer; this was supposed to be caused by the valve area being too small to allow the pump cylinder to fill with water before the completion of the stroke.

Tracings of indicator cards taken at different speeds and pressures are given. Cards Nos. 9 and 10 were taken before alterations were made in the main steam valve; card No. 9 before adding the distance pieces on the ends of the main valve, and card No. 10 before placing the taper wedges in the steam grooves in the steam sleeve.

APPENDIX.

CALCULATIONS.

Columns.

1. From data.
2. From data.
3. From data.
4. From data taken at beginning of each test, and corrected by Smithsonian Meteorological Tables.
5. From data, average of fifteen-minute readings.
6. From data, average of fifteen-minute readings.
7. Column 6 plus barometer pressure in pounds per square inch.
8. From data, average of fifteen-minute readings.
9. Average from indicator cards taken at fifteen-minute intervals.
10. Average from continuous counter.
11. $\frac{\text{Column 26}}{2 \times 60 \times \text{Column 25}}$.
12. Average from indicator cards taken at fifteen-minute intervals.
13. As weighed.
14. Calculated from data, from throttling calorimeter,

$x =$ $[H - q + S\ (T_1 - Tc)] \div r$.

$r =$ latent heat, due to pressure in steam pipe.

$q =$ heat of liquid, due to pressure in steam pipe.

$H =$ total heat in one pound of dry steam at calorimeter pressure.

$Tc =$ Normal temperature of steam in calorimeter, due to calorimeter pressure.

$T_1 =$ Reading of thermometer in calorimeter.

$S =$ Specific heat of superheated steam at calorimeter pressure.

15. Column 13 × Column 14.

16. Column 15 ÷ 60.

17. Column 15 × total heat of steam corresponding to pressure given in Column 7.

18. Assuming the quality of the exhaust steam to be 85 per cent. dry and used in a feed-water heater of 90 per cent. efficiency; then this column would be obtained:

$$[.85\ r + q] \times \text{column } 15 \times .9.$$

$q =$ heat of the liquid corresponding to pressure in column 8.

$r =$ latent heat of steam corresponding to pressure in column 8.

19. Column 17 — Column 18.

20. From data, average of fifteen-minute readings.

21. From data, average of fifteen-minute readings.

22. Column 21 ÷ 2.035.

23. From measurement.

24. Column 20 — Column 22 + Column 23.

25. Average from total readings of revolution counter.

26. From continuous-stroke recorder.

27. Tests 5 to 16 and 21 to 32, inclusive, from venturi-meter readings; all other tests from calibrated tank.

28. $A \times L \times D$; where $A =$ area of plunger minus one half the area of the rod, in square feet; $L =$ total travel of piston in feet per hour (Column 16); $D =$ density of water in pounds per cubic foot at the temperature given in column 30.

29. Column 28 — Column 27.

30. From data, average of fifteen-minute readings.

31. Duty $= \dfrac{H \times W \times 1{,}000{,}000}{h}$, where:

$H =$ total head in feet = Column 24 × 2.306.

$W =$ weight of water pumped = Column 27.

$h =$ total number of heat units consumed by pump (Column 19).

32. $\dfrac{\text{Column 28} - \text{Column 27}}{\text{Column 28}} \times 100.$

33. $\dfrac{A \times L \times 7.4805}{144}$.

$A =$ area in square inches of pump plunger minus one-half the area of the rod.

$L =$ total travel of pump plunger in feet per hour, Column 26.

34. Column 33 × $\dfrac{100 - \text{Column 32}}{100}$.

35. $\dfrac{\dfrac{H \times W}{60 \times 33{,}000}}{\text{I.H.P.}} \times 100.$

I.H.P. from Column 12.

H = total head in feet, Column 24 × 2.306.

W = weight of water pumped, Column 27.

36. Column 15 ÷ Column 12.

37. $\dfrac{\text{Column 15} \times 1{,}000}{\text{Column 27}}$.

38. Assuming that the feed water is heated to 200 degrees F. by the exhaust from the pump, then the total B.T.U.s absorbed by the pump will be:

Col. 13 [(Col. 14 × r + q) − (200–32)].

where r = the latent heat of steam at pressure (Column 7), and

q = heat of liquid at the same pressure.

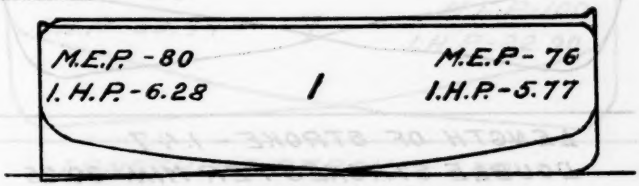
39. Duty = $\dfrac{H \times W \times 1{,}000{,}000}{h}$, when

H = total head in feet = Column 24 × 2.306.

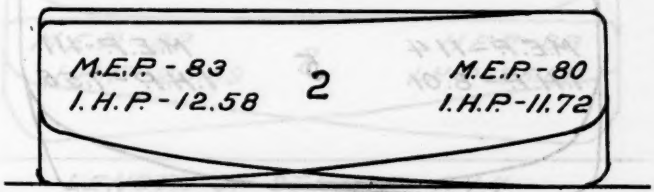
W = weight of water pumped = Column 27.

h = total number of heat units, consumed by pump, Column 38.

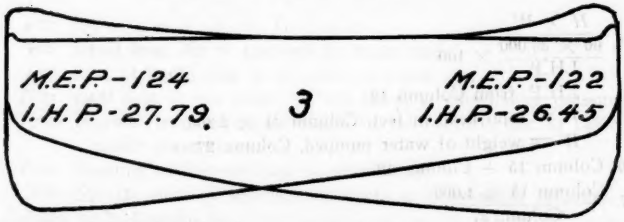
NOTE.—Marks and Davis' steam tables were used in making all steam calculations.



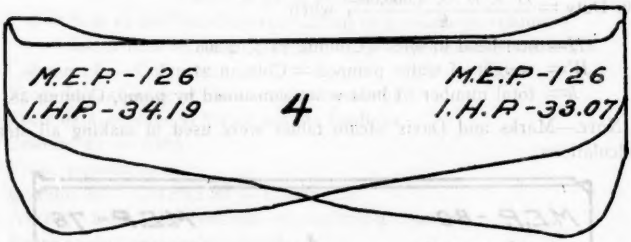
LENGTH OF STROKE — 1.27
DOUBLE STROKES PER MIN. - 10.13
DISCHARGE PRESSURE - 200
SCALE OF SPRING - 100



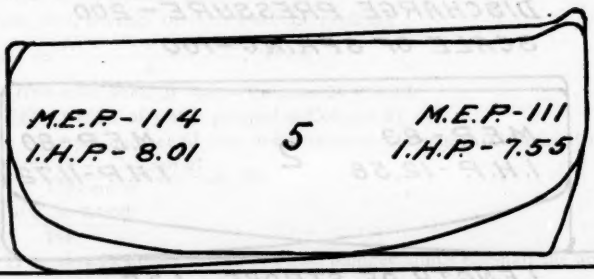
LENGTH OF STROKE - 1.35
DOUBLE STROKES PER MIN. - 18.4
DISCHARGE PRESSURE - 200
SCALE OF SPRING - 100



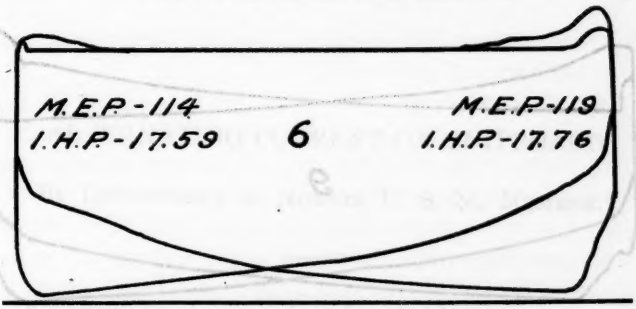
LENGTH OF STROKE - 1.41
DOUBLE STROKES PER MIN.-26.06
DISCHARGE PRESSURE-300
SCALE OF SPRING-150



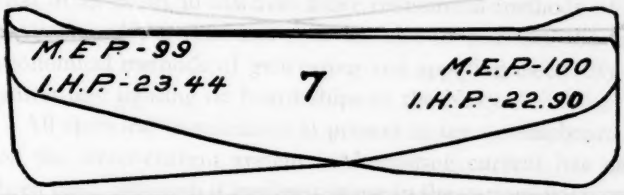
LENGTH OF STROKE - 1.47
DOUBLE STROKES PER MIN.-30.26
DISCHARGE PRESSURE-300
SCALE OF SPRING-150



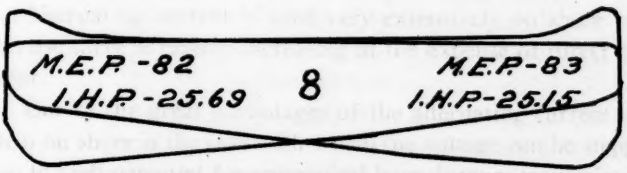
LENGTH OF STROKE - 1.34
DOUBLE STROKES PER MIN. - 8.6
DISCHARGE PRESSURE-300
SCALE OF SPRING-100



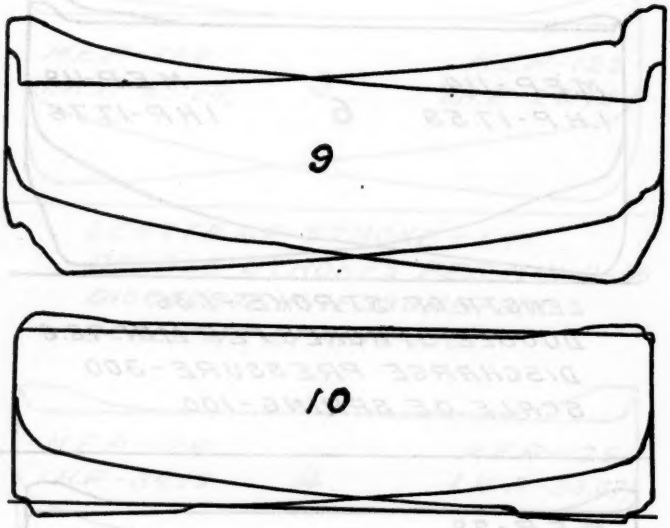
LENGTH OF STROKE - 1.36
DOUBLE STROKES PER MIN.-18.6
DISCHARGE PRESSURE-300
SCALE OF SPRING-100



LENGTH OF STROKE - 1.40
DOUBLE STROKES PER MIN.-27.73
DISCHARGE PRESSURE-250
SCALE OF SPRING-150



LENGTH OF STROKE - 1.44
DOUBLE STROKES PER MIN.-35.66
DISCHARGE PRESSURE-200
SCALE OF SPRING-150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ALTERNATING CURRENT ON SHIPBOARD.

By Lieutenant A. Norris, U. S. N., Member.*

Electricity has been used on board ships of the Navy for lighting and auxiliary purposes for many years, but the question of economy in its use and application has had comparatively little consideration. The competition which has been inaugurated in the engineering branches of the service has, however, caused many subjects to be taken up for investigation in an effort to discover more economical methods of operation, and the time has now arrived to consider the most economical methods of generating and applying electricity for power and lighting on board ships of the Navy.

All electrical installations at present in use on shipboard are on the direct-current system. Alternating current has never been used, although it has been in use in the various navy yards and commercially for a number of years. It is claimed by many that alternating current cannot be successfully applied to the many and varied uses for which electricity is required on naval vessels, for the reason that, for most purposes, it is not as flexible as the direct-current system at present installed.

Alternating current is used very extensively on shore. and its use there is rapidly increasing at the expense of direct current.

One of the great advantages of the alternating-current system on shore is the ease with which the voltage can be stepped up to high potential for economical long-distance transmission, and then stepped down to low voltage for commercial use at any place along the transmission line. This principle of volt-

* Lieutenant (J. G.) V. V. Woodward, U. S. N., assisted in the preparation of this paper.

age transformation holds to some extent on board ship, as will be shown later.

Since the introduction of the steam turbine, which is essentially a high-speed machine, practically all electrical power in power plants is generated on the alternating-current system, because the steam economy of the steam turbine can only be obtained at high rotative speeds. On the other hand, the direct-current generator is inherently a comparatively low-speed machine, and if run at the high economical rotative speed of the steam turbine a prohibitive violent sparking of the commutator would result.

All early electrical apparatus was built on the direct-current system, consequently the direct-current system has been developed to a high degree, but during the last few years, since the steam turbine has directed study along alternating-current lines, vast strides have been made and are continuing to be made in the application of the alternating-current system to all commercial purposes.

Considerations which have led to the adopting of the alternating-current system on shore are:

In the power plant:

I. Cheaper first cost of installation.
II. Cheaper cost of production per kilowatt hour.
III. Cheaper mechanical cost of upkeep.
IV. Cheaper cost of labor upkeep.
V. Less space required for generators.
VI. Less weight of generators.
VII. Higher generated potential.
VIII. Simpler construction of generators.
IX. Higher efficiency of generators.
X. Increased ease of voltage regulation.

In transmission of power:

I. Ease of stepping up and down potential.
II. Ease of transmission at high potential.
III. Saving in cost of transmission due to high potential.

IV. Saving of weight in transmission lines.
V. Reduction of losses in transmission lines.

Application (motors):

I. Cheaper motors.
II. Simpler construction of motors.
III. Less weight of motors.
IV. Less space occupied by motors.
V. Less cost of upkeep and repair of motors.
VI. Use of open motors where closed are now necessary.

The electrical installation of a battleship is such a complicated problem that it must be considered in detail to determine whether the alternating-current system will be satisfactory in all respects. In order to do this it will be necessary to enumerate the electrical appliances used on board ship.

These are on board a modern battleship—

¾–H.P. motors—long-arm system.
35–H.P. motors—turret turning.
50–H.P. motors—cranes, hoisting.
30–H.P. motors—cranes, rotation.
35–H.P. motors—deck winches.
3–H.P. motors—ammunition hoists.
3–H.P. motors—fresh-water pumps.
12–H.P. motor—sanitary pump.
6–H.P. motor—laundry.
4–H.P. motors—ammunition hoists.
40–H.P. motors—turret hoists.
10–H.P. motors—turret hoists.
15–H.P. motors—turret-gun elevating.
10–H.P. motors—turret rammers.
5–H.P. motors—turbine jacking.
45–H.P. motors—forced draft.
1¾ to 11–H.P. motors—ventilating sets.
¼ to 2–H.P. motors—galley.
¾ to 7½–H. P. motors—machine shop.
Motors—12-inch bracket fans.
Motors—16-inch bracket fans.

500-watt motor generators—interior communication.
25-watt dynamotors—telephone bells.
1,500-watt motor generators—turret-danger zone
5,000-watt motor generators—wireless.
Small motor generators—howlers.
100-kilowatt ranges.
33-kilowatt bake ovens.
15-kilowatt searchlights.
25 to 500-watt incandescent lamps.
15 to 500-watt electric heaters.
12 mercury-vapor lamps.
165-H.P. motor—steering engine.

In considering the installation of an alternating-current system on shipboard the principal factor to be determined is the adaptability of the alternating-current motor to the various uses for which electric power is required.

The following are the forms of alternating-current motors now manufactured:

I. Synchronous motors.
II. Series-commutator motors.
III. Induction motors.
IV. Repulsion motors.

The synchronous motor is inherently a constant-speed motor and could be used only for that purpose. It possesses but poor starting torque, readily breaks down when subjected to widely or rapidly fluctuating loads. Direct-current field excitation is also required, which is a great disadvantage. The synchronous motor is seldom manufactured in small units. From these considerations it is not considered a suitable motor for shipboard use.

The alternating-current series type has recently come into use, but is principally employed in single-phase traction systems, and it is not suitable in its present development for use on shipboard, due to its excessive weight, poor efficiency, low power factor and serious commutator troubles.

The induction motor is the simplest motor manufactured

and appears to meet the requirements of ship use. It will be described in detail.

There are several forms of repulsion motor manufactured, but all are unsuitable for naval use, due to the use of a commutator and serious sparking troubles.

The alternating-current induction-type of motor will now be considered, it being the best motor for the various duties required aboard ship. It is of most rugged and simple construction; there is no commutator, the source of so much trouble in direct-current machines. Freedom from sparking adapts this motor for use in any part of the ship, such as the handling rooms, magazines and ammunition passages. If the stator windings are properly insulated the motor can be run under water as readily as in the air. There is no leakage whatsoever, even in salt water, in spite of there being bare copper conductors of the rotor in actual contact with the water. It requires little attention other than proper lubrication and it has proved itself a most reliable piece of machinery.

INDUCTION MOTORS ON SHIPBOARD.

The successful application of the induction motor on shipboard requires a knowledge of the operating characteristics of this motor, and also those of the apparatus to be driven, so that the form of motor installed may be in every way suitable for the work it has to perform. Before it is possible to determine the most suitable type for any particular service, it is necessary to make a complete series of characteristic curves, and from these, data relating to (1) variations of speed and torque, (2) slip, (3) rating, (4) efficiency and power factor, may be obtained. These characteristics have a direct bearing on the performance of a motor, and they may be considered as follows:

Speed-Torque Characteristics.—The most important characteristic of the induction motor is that which shows the relation between the torque and the speed at which it is developed. When load is put on a motor its speed falls off until the in-

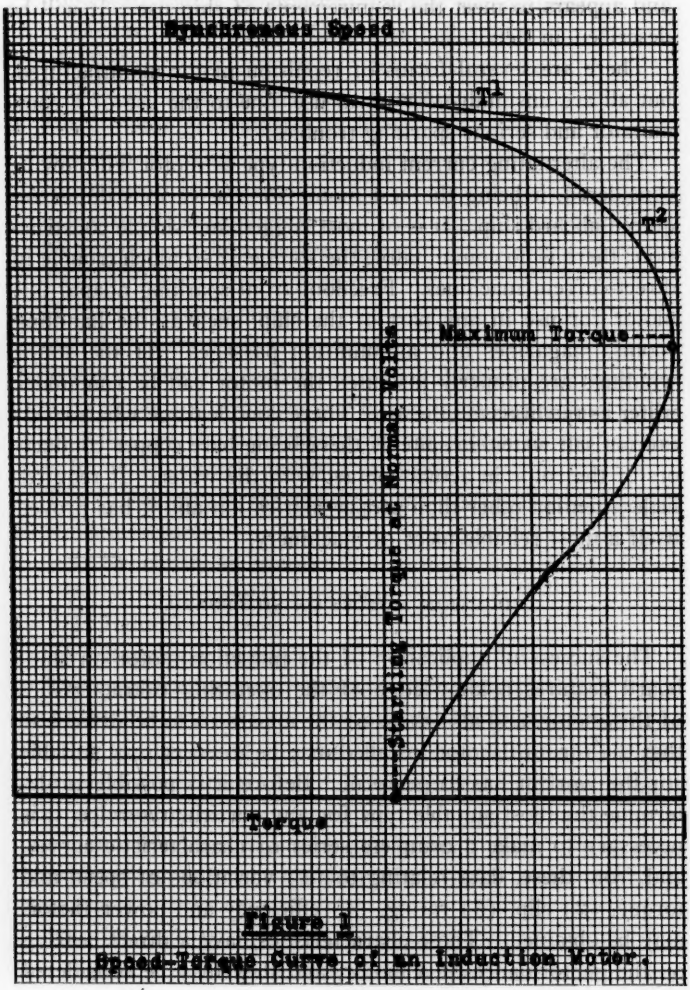


Figure 1.
Speed-Torque Curve of an Induction Motor.

duced E.M.F. in the secondary is sufficient to develop the torque required by that particular load. As the load is increased the torque increases until the stalling point (pull-out torque) is reached, when both torque and speed fall off rapidly

until the motor stops. If the increase in torque were proportional to the decrease in speed the resultant speed-torque curve would be a straight line, as represented by T^1 in figure 1. In practice, however, a speed-torque curve assumes the form of T^2, for the following reasons:

(1) Since the counter electromotive force of the primary winding decreases as its IR drop increases, the magnetic field will be diminished, so that the speed must fall off more rapidly than the torque increases to generate the induced currents required in the secondary.

(2) The induced electromotive forces in the secondary are opposed by a counter electromotive force, consisting of a series of local electromotive forces set up by the stray magnetic fields, due to the primary and secondary currents, which necessitates a further drop in speed for any particular value of torque.

If the speed of an induction motor remains constant the torque developed would be proportional to the watts output and the losses in the secondary, but, as before stated, the torque increases to a greater extent, owing to the fact that the speed drops as the load increases.

The torque developed is directly proportional to the watts input to the secondary and varies as the square of the voltage impressed on the primary winding, owing to the fact that an increase in the voltage causes a decrease in the leakage. As a motor requires a rotor of fairly high resistance to enable it to exert a large torque at starting, it will be of interest to consider the effect of varying values of resistance in the secondary circuit, reference being made to the curves in figure 2, which represent the torque at different speeds corresponding to different values of controller resistance in the rotor circuit of a variable-speed motor. From these curves it will be seen that as the secondary resistance increases, the slip is increased proportionally if the same value of torque is required; also, a low secondary resistance gives the smallest drop in speed for a given increase in the torque developed, but gives a low starting torque at the expense of a large starting current, so that

this condition is, therefore, not suitable for running purposes. When the slip is large, the starting torque is greatly increased for a small increase in the starting current, but falls off rapidly as the speed approaches its synchronous value, the torque developed at any given "slip" being proportional to the square of the impressed voltage. It will, therefore, be seen that in-

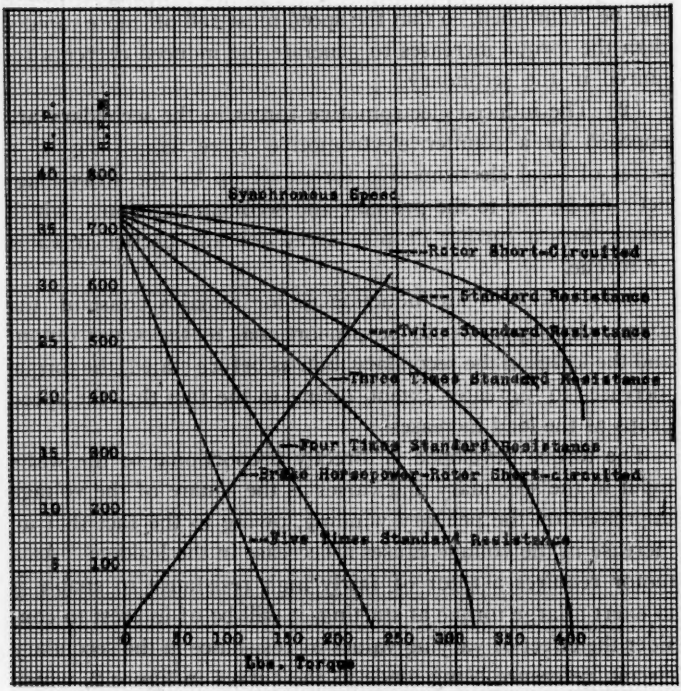


Fig. 2.—Speed–Torque Curve Corresponding to Differential Values of Resistance in the Secondary Circuit.

creasing the rotor resistance within certain limits will cause an increase in the "slip" and "starting torque," and diminish the starting current; also, the torque is directly proportional to the rotor resistance at starting, but "inversely" when the motor is running on load.

Slip.—As is well known, the synchronous speed of an induction motor is given by the number of alterations of the supply circuit, divided by the number of pairs of poles, and the difference between this speed and the actual speed at any particular load represents the "slip," which is generally expressed as a percentage of the synchronous speed. The slip increases with the load, so that the torque developed is increased sufficiently to take charge of that particular load. The slip is directly proportional to the resistance of the rotor winding; it also represents the I^2R loss in the winding, because the slip at any particular load is just sufficient to generate the induced electromotive forces necessary to drive the rotor current through the ohmic resistance of the windings. For these reasons the "slip" expressed as a percentage of the synchronous speed is equal to the I^2R loss in the rotor expressed as a percentage of the watts input to this winding.

The slip varies inversely as the square of the impressed voltage; that is to say, with constant torque, if the voltage is doubled the "slip" is reduced to one-quarter of its former value.

Rating.—A motor is generally rated so that it will carry its normal load continuously with a definite rise in temperature, which is usually about 50 degrees centigrade. The rating of a motor should depend upon its type and also on the conditions under which it has to operate. When a motor has to operate under continuous running conditions, the foregoing is the only satisfactory rating; but in the case of deck winches or cranes, where the maximum power is required only occasionally and for short periods, the rating should be governed by an "intermittency factor." With this method of rating, although a motor may be considerably overloaded for short periods, at frequent intervals it will be idle, and so have every chance to cool off. When the rating of a multi-speed motor is to be governed by an intermittency factor, the fact that it has sometimes to deliver its maximum power at its lowest speed should not be overlooked. Under these operating conditions the

effect of an intermittency factor should not be so great as in the case of the constant-speed machine.

An intermittency factor affects the heating limit on which depends the rating of a motor, and, as a machine can stand a larger overload when running at its highest speed, owing to improved ventilation, too much allowance should not be made for intermittent working when a motor is called upon to deliver heavy overloads at low speeds, from the point of view of overheating.

Efficiency.—The ratio of the actual to the synchronous speed of an induction motor is sometimes taken as a measure of its efficiency, which is inversely proportional to the resistance of the rotor winding, consequently it is inversely proportional to the slip. An induction motor generally has its maximum efficiency at about three-quarters load, which indicates the most suitable distribution of the losses for obtaining a motor capable of good all around performance. There is a large variation of the efficiency at different speeds for any given value of torque, and as an increased torque is accompanied by increased rotor losses, it is impossible to have large torque and high efficiency. The fact that a motor has low efficiency at low speed is not of great importance, unless it is required to run at reduced speed for considerable periods.

Power Factor.—The power factor of an induction motor is low at light loads but increases rapidly with the load; when a motor is working much below normal voltage its power factor will be correspondingly low. On the other hand, an increase in the voltage above normal value will also cause a decrease in the power factor, which is due to increased iron losses. The iron loss is made up of magnetizing and leakage components, and as the magnetizing component has a constant value at all loads and the value of the leakage component varies with the load, the effect of the former on the power factor will be greater at light loads, consequently a motor having a large magnetizing and small leakage current will have a low power factor at light loads. A small magnetizing and a large leakage current will produce a high power factor at light loads, which

will fall off rapidly as the load increases. When the motor is required to develop a high maximum torque, as this requires a large magnetizing current, the power factor at light loads will be low. In cases where the magnetizing component is small compared with the leakage component the power factor at light loads will be high and the maximum torque will be low.

The characteristics which have been considered are common to all induction motors and have a direct bearing on their application to ship use; but before considering the selection of the most suitable type of motor for any particular service it will be necessary to make a comparison of the characteristics of the various types, and for this purpose the induction motor may be divided into two distinct classes, viz: (1) constant-speed motors and (2) variable-speed motors. These may be again subdivided into two types, depending upon the nature of their rotor windings, (1) squirrel-cage or permanently short-circuited rotor; (2) phase or polar-wound rotor with connections to some external resistance, each of these types being suitable either for constant or for variable-speed service.

CONSTANT–SPEED MOTORS.

This class of motor has the smaller slip and also good speed regulation, running at almost constant speed from no load to heavy overloads. The rotor is usually of the squirrel-cage type when the rating of the motor does not require a large starting current and frequent starting is not necessary. As the slip is very small, this type has high efficiency and excellent speed regulation. The characteristics of this type are illustrated by the curves in figure 3, taken from a test sheet, which shows the performance of a seventy-five horsepower, three-phase induction motor. It will be seen that the full-load values are:—slip, 4 per cent.; efficiency, 87.5 per cent.; and power factor, 89.5 per cent. The power factor increases with heavy overloads on the motor. The full-load torque is approximately 680 pounds, and the torque exerted at normal

voltage (200 volts) with the rotor locked, was 2,225 pounds, so that the starting torque is 2,225 ÷ 680 or 3.27 times the value of its full-load torque.

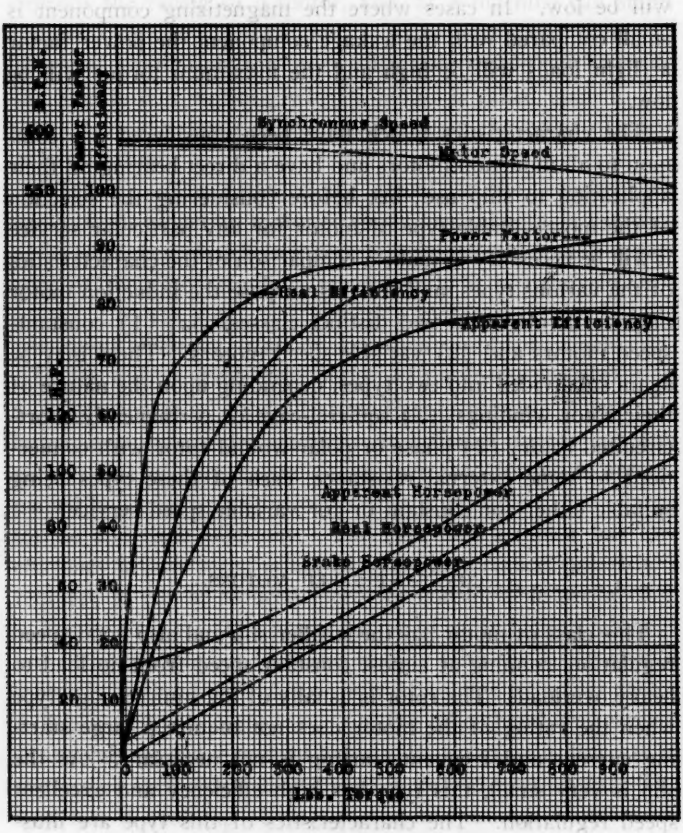


Fig. 3.—Characteristics of Induction Motor.

The starting torque is much less than the maximum torque the motor will develop before stalling or the pull-out torque is reached. It is impossible to obtain a large starting torque with small starting current from this type of motor, as this requires a large slip, but the starting conditions may be im-

proved at the expense of efficiency by substituting short-circuiting rings made from a material having a higher resistance and smaller section.

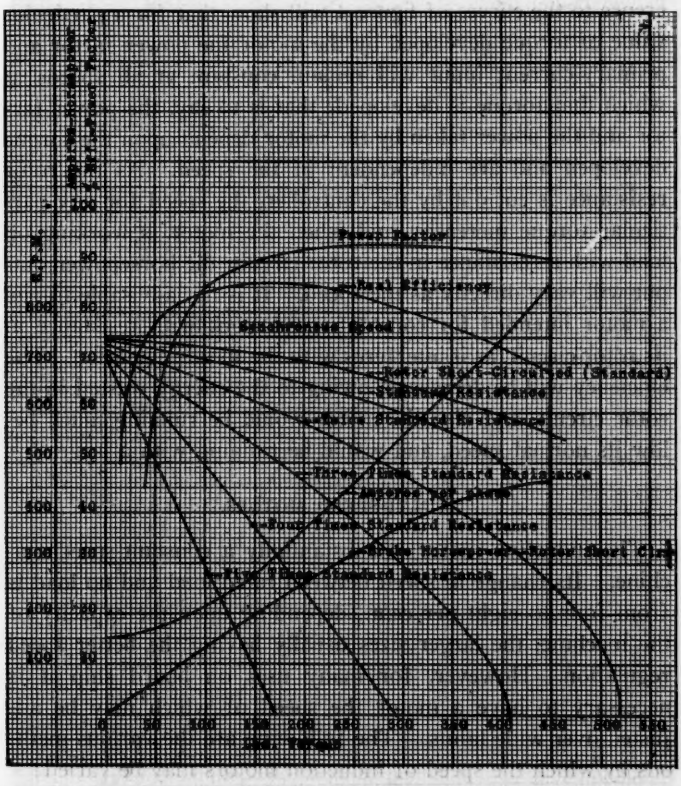


Fig. 4.—Speed–Torque Curve of Polar–Wound Motor for Various External Resista. ces in its Rotor Circuit.

Polar Wound Rotors.—The employment of a motor having a wound rotor in conjunction with a variable resistance in the rotor circuit gives a high-resistance rotor at starting and a low-resistance rotor when running on load, which means high starting torque with low starting current. The speed-torque curves obtained from a motor of this type for varying values

of external resistance in its rotor circuit are shown in figure 4, from which it will be seen that the speed remains nearly constant at all loads when the rotor is short circuited; the motor will also operate at high efficiency and power factor. A reference to the curves of figure 4 will show that for a particular value of torque developed by the motor the power factor and primary-current input will remain constant at all speeds at which that torque is developed; that is to say, the power factor and the power taken by the motor depend upon the torque developed and are independent of the speed. As the rotor resistance is cut out by the controller, the speed for any particular torque increases, and when the motor is working on load with a resistance in the rotor circuit, the horsepower output will vary directly as the speed. It is not advisable to run a motor intended for constant-speed work with resistance in its rotor circuit, as it interferes with the speed regulation. The efficiency will be decreased, owing to the energy wasted in the rotor circuit, and also the motor would be running much below its normal speed, and would be running with a large slip.

MULTI-SPEED MOTORS.

The induction motor is substantially a constant-speed machine. Its change in speed between rated load and no load is from four to eight per cent., depending upon the capacity of the machine, the larger sizes usually having the better speed regulation. However, for many uses on board ship it is necessary to have a motor the speed of which may be adjusted over wide limits. The following are the various methods by which the speed of induction motors may be varied:

I. Variation of applied potential.
II. Variation of the rotor resistance.
III. Variation of the number of motor poles.
IV. Variation of the frequency of the supplied voltage.
V. Cascade or concatenated connections.

Methods I, II and III, are the only ones applicable for use on shipboard; the others are too complicated.

Speed Control by Variation of Applied Potential.—The slip

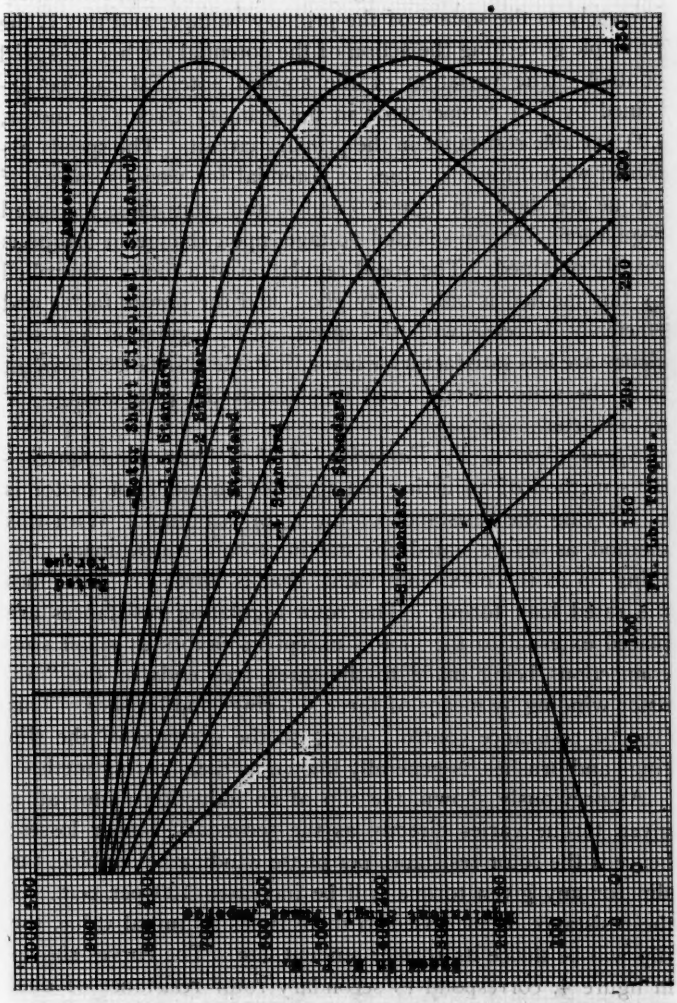


Fig. 5.—Speed-Torque Curve with Various Values of Rotor Resistance.

of an induction motor at a given torque varies approximately inversely as the square of the primary voltage, and this is the principle governing the action of the potential method of speed control. The variation of the applied potential can be obtained by (1) compensator in the primary circuit of motor, (2) variable resistance in the primary circuit. The speeds obtained with different values of potential for various values of torque are shown in figures 6 and 7, and from these the speed regulation, for any selected voltage, upon change in torque is readily obtained. The speed regulation of a motor controlled by this method is only fairly good, and the power factor and efficiency decrease with the speed. The regulation and efficiency are less satisfactory when adjustable resistance in circuit with the stator is employed in place of the compensator or auto-starter.

Speed Control by Variation of the Rotor Resistance.—This method of adjusting the speed of an induction motor is by varying the resistance of the rotor windings. With this method, as before stated, the starting torque can be made anything up to the maximum value, that is, three or four times the rated load torque. The rotor-regulating resistances are placed external to the motor, except in motors of small size, connections being made to the three free ends of the "Y" rotor winding by means of three slip rings and brushes. This type of resistance control, owing to the presence of the slip rings and brushes, is generally known as the slip-ring method.

As previously shown, the slip of an induction motor at a given torque varies directly as the secondary copper losses; hence if the rotor resistance per phase winding be doubled, the slip for any given torque will be increased 100 per cent.; if the resistance be increased to three times its initial value, the slip will be thrice its former amount, etc. The curves shown in figure 4, correspond to secondary rotor resistance of one, two, three, four, five and six times that existing with the rotor short circuited, and the curves in figure 5, to one, one and one-half, two, four, five and eight times that existing with the rotor short circuited. These externally added resistances

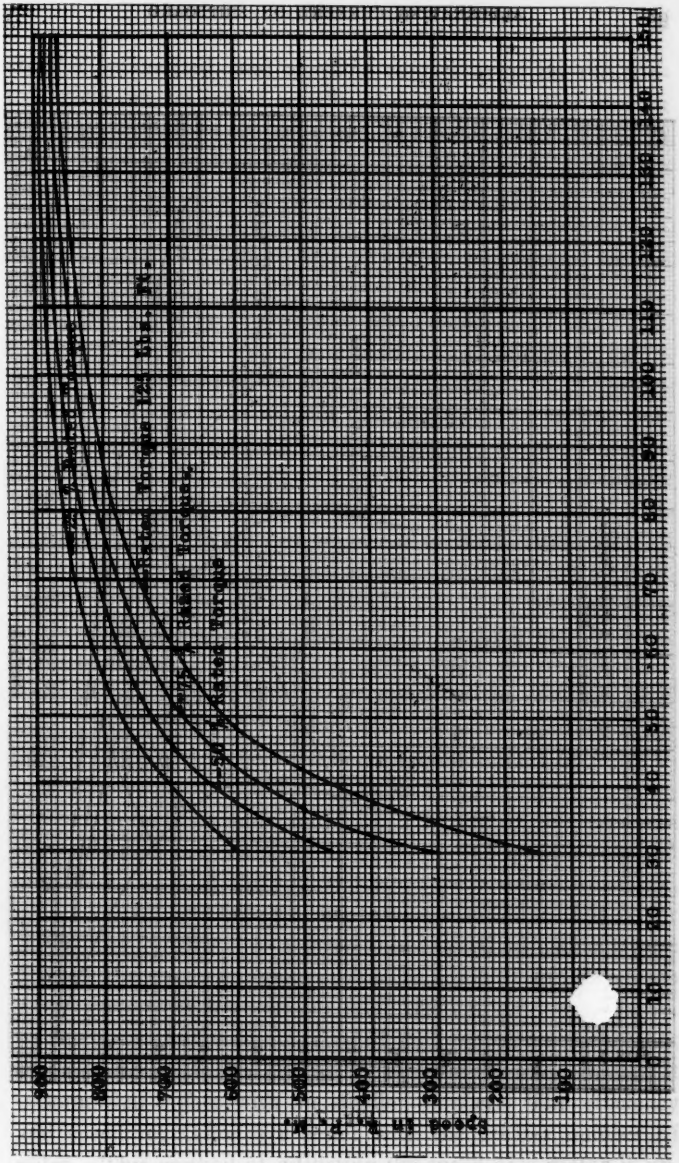


Fig. 6.—Speed–Voltage Curves of a Three-Phase 20-H.P. Induction Motor.

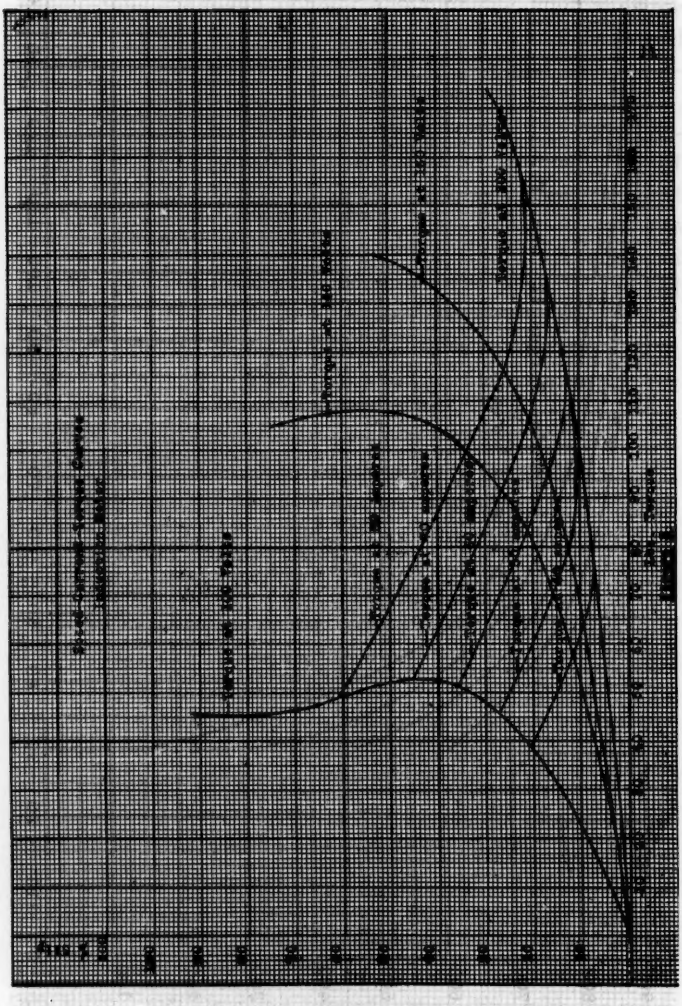


Fig. 7.

are "Y"-connected, and movable contact arms on the controller cut-out resistance equally in each phase. The amount of resistance left in the rotor circuit would depend upon the speed required, and it is possible so to proportion it that the motor would develop the full-load torque at starting with little more than full-load current. The speed varies automatically with the torque and horsepower developed by the motor.

Speed Control by Variation of the Number of Motor Poles.—The synchronous speed in revolutions per minute of an induction motor being given by the expression 60 × frequency ÷ pairs of poles, it is evident that the speed varies inversely as the number of poles. Thus a motor wound for six pairs of poles and operating normally at 600 revolutions per minute will rotate at a speed of 1,200 revolutions per minute if its stator winding be rearranged so as to have three pairs of poles. The simplest method of applying this control is to employ a stator having two or more windings, corresponding to the different number of poles. One winding may be used, it being arranged for different speeds by means of a commutator switch, which alters the grouping of the coils and thus the number of poles. A rotor of the squirrel-cage type is the only practicable one, because, being short circuited upon itself, it is adapted to any number of poles.

The connections of a multi-speed motor of this type are relatively simple, especially if only two-to-one speed is required and the rotor is of the squirrel-cage type. In such instance, only six leads are brought out from the machine if it is for three-phase circuits. If a three-to-one speed adjustment were wanted, three pole groupings would be required and nine leads would have to be brought out from the stator.

The power factor of this type of multi-speed machine is not greatly affected by change in the number of poles, though it is somewhat higher with the smaller number. The efficiency and speed regulation are better with the greater number of poles. This method of adjusting the speed of a motor is undoubtedly the most satisfactory as regards efficiency and ex-

cellence of speed regulation. High first cost and the necessity of disconnecting it from the line to change the speed are the only objections.

SUMMARY.

In general, motors may be divided into the following classes:

(1) Those not requiring large starting torque nor variable speed, but requiring good speed regulation with change of load.

(2) Those requiring good starting torque, but not requiring variable speed.

(3) Those requiring considerable starting torque and a speed falling off considerably with increase of load.

(4) Those requiring variable speed, either large or small torque and reversibility.

The requirements of these classes are fulfilled, respectively, by induction motors of the following tubes:

Class I, by an induction motor with a low-resistance squirrel-cage rotor.

Class II, by one with a medium to high-resistance squirrel-cage rotor.

Class III, by a motor with polar-wound rotor connected to external resistance.

Class IV, by one of the following ways:

(a) Variable resistance phase wound rotor, gives variable starting torque, and also variation of speed depending upon the amount of resistance in circuit with the rotor.

(b) Variable resistance in circuit with the stator winding gives variable speed with small variable starting torque.

(c) Variable speed with constant torque at each speed is obtained by changing the number of motor poles.

From a consideration of the above summary it will later be shown that all of the conditions met with on board ship may be satisfactorily fulfilled by the induction motor in one of its various forms.

NAVAL ENGINEERS

APPLICATION OF ALTERNATING CURRENT.

In proposing an alternating-current installation for a battleship it has been attempted to meet, as far as practicable, all the requirements of power and lighting as are now demanded by the electrical specifications for vessels of the Navy.

A voltage of 230, 60-cycle, three-phase, will be used for the generators, and 220 volts for the power receivers. This drop of voltage between generator and receiver will conform to present specifications as regards line drop.

The use of the higher voltage is in line with good commercial practice and gives a saving in weight and cost of material. Any prejudice against the increased voltage would seem to be the result of conservatism on the part of the service at large. Similar voltages have been used on naval vessels abroad for some years, and this should tend to allay any fear of liability to dangerous shock in the handling of this higher potential. It is believed that 230 volts alternating current will not be in the least injurious to the personnel, and the small discomfort resulting should be of small weight when compared with the great advantages to be derived. Besides the adoption of a voltage of 220 by several foreign powers there is one instance recorded of the use of 500 volts on a foreign merchant vessel, and which resulted in the saving of $33,000.00 in the first cost of installation. This, however, in view of the confined spaces aboard ship and the possibility of leakage, is a very high voltage to use, in fact, it seems prohibitive on board a man-of-war.

The electrical system aboard ship is a combined lighting and power system, and consequently the same frequency would have to be used for both purposes. Twenty-five cycles would be the best for motors, but such a frequency is too low for lighting purposes, so 60 cycles has been chosen, in accordance with the best commercial practice. The heating requirements of the naval specifications are much more severe than in regular commercial practice, and as a result the greater part of the electrical machinery on board ship must be of special design.

In proposing the alternating-current system we have endeavored to retain the standard Navy heating requirements for all electrical appliances, but in comparing the alternating-current system with the present direct-current installation, the fact that all alternating-current machinery is of commercial type must be considered.

In contemplating any alternating-current installation the power factor of each individual receiver must be considered in order to properly design the size of the wiring and the capacity of the generating plant.

The lighting load may be considered as practically non-inductive, as the power factor will average about 96 per cent. The motor-load power factor, considering that all motors are of the induction type, can be taken to lie somewhere between 75 and 90 per cent. We have adopted as the power factor under battle conditions for the entire plant an average of about 80 per cent., this being a very conservative estimate and close enough for present purposes.

GENERATING PLANT.

The generating plant as designed for a modern battleship consists of four 300-kilowatt generators, direct-current, multipolar, compound-wound, to run at a constant speed of 1,500 revolutions per minute, and to deliver 125 volts at the terminals at rated speed with load varying between no load and one and one-third rated load.

In place of this plant it is proposed to substitute the following turbo-generators: three 500-k.v.a. units, alternating-current, three-phase, sixty-cycle, 3,600 revolutions per minute. The increased output of the plant is necessary in order to allow for the assumed power factor of .80.

For the field excitation of these generators there will be required two 45-kilowatt, 125-volt, direct-current, turbo-generators. One of these machines is sufficient for the excitation of the fields of the three generators at full load, the second machine being installed as an emergency precaution. The ex-

citers are shunt wound and the field excitation is to be controlled by a Tirrill regulator so as to obtain constant terminal voltage of the main generators.

COMPARISON OF WEIGHTS AND OF WATER RATES.

The following table will give a comparison of the weight, water rates, etc., of the present direct-current machines and the proposed alternating-current installation.

	No. of Gens.	Capacity.	Weight.	Water rate.	R.P.M.
Direct Current:	4	1,200 kw.	128,000	Full L. 25.5	1,500
				Half L. 31.0	
Alternating Current:	3	1,500 kw.	102,000	Full L. 18.5	3,600
				Half L. 20.25	
Exciter sets:	2	90 kw.	16,000	Full L. 33.0	3,600
				Half L. 45.0	

The total saving in weight in favor of the alternating-current installation is 10,000 pounds.

The economy of operation for the alternating-current installation can be worked out as follows: Assuming that the full-load excitation requires 15 kilowatts for each main generator we have $15 \times 33 = 495$ pounds of water per hour per each main generator; therefore we must add $495 \div 500$ pounds of water to the full-load water rate of the main generators as given in the preceding table to get the true water rate for comparison.

This factor added to the water rate as given above will give an alternating-current water rate of 19.5 pounds per k.v.a. hour at full load. If we consider the power factor of the proposed system to be .80 the above water rate would have to be very slightly increased to compensate for the internal generator and line losses due to the wattless component of the generated current. This increase would be so infinitesimal as to be practically negligible in an alternating-current ship installation.

On comparison with the direct-current water rate we find a saving of 23.5 per cent. in water consumption over the direct-current installation. This gain for the alternating-current installation is due to the increased efficiency of the generator and to the greater economy of the high-speed turbine.

OTHER ADVANTAGES.

The present direct-current installation requires frequent overhauling of the commutators. In the alternating current installation this trouble is eliminated except in the small exciter sets. This is an appreciated advantage, as it is a well known fact that the failing of the commutator under heavy overloads has been the cause of the loss of many generators when their services have been most needed.

Overload conditions of the generator plant are frequent aboard ship. These conditions can be handled far more satisfactorily and with less danger by the alternating-current generator than by the present direct-current machine.

The manipulation of the alternating-current switchboard in alternating-current parallel operation requires better training and more attention on part of the personnel, but the units themselves require much less attention in operation and upkeep.

SWITCHBOARDS, DISTRIBUTION BOARDS, ETC.

The specifications for a battleship require a generator switchboard in each dynamo room for the control of the generating units, also two bus-feeder panels. From these panels separate lighting and power bus feeders run to distribution boards, two in number, one aft to supply the after portion of the ship, and the other forward to supply the forward portion of the ship. The bus feeders to the distribution boards are so arranged that either board may be energized from the other. These bus feeders are run in wiring passageways beneath the protective deck.

The present system of distribution will be followed in the proposed alternating-current installation. The cost of and the

space required by the two systems will be practically the same except that, due to the higher voltage and the three-phase alternating current, all power buses will require much less copper, although an allowance must be made for operating the system with a power factor of .80.

The number of instruments in the alternating-current installation will be greater and the weight in this regard will be higher.

WIRING.

The present instructions for battleships require part of the wiring to be in metallic conduit, part to be protected by metallic wiring passageways, and part is entirely open and unprotected. For the purpose of reducing weights the present specifications require as little conduit to be used as possible.

In the alternating-current installation it is considered necessary to run all three wires of the cable in the same conduit in order to prevent, as far as possible, induction in telephone and other interior-communication circuits and to minimize the effect of self inductance. If all cable is run in conduit it is believed that no trouble whatever will be experienced from induction.

LIGHTING WIRING.

The present lighting wiring is on the two-wire system operating on 125 volts at the distribution boards. In the alternating-current system it is the intention to use the same system both as to distribution and as to voltage. In order to carry out this scheme of lighting we propose to use two three-phase transformers on the lighting system to reduce the generator voltage down to 125 volts at the distribution boards. The battle-load and cruising-load wiring will be run on the two-wire system as at present, but so distributed from the distribution boards that approximately one-third of the load will be taken by each phase. This will practically balance the lighting load on the generators; any unbalancing will be very small and of no consequence.

It is considered necessary to reduce the voltage for lighting purposes from the generated voltage to 125 volts for the reason that all incandescent lamps of 220 volts are very unreliable both as to power and life. With the exception of the addition of the transformers the lighting installation will be the same as the present direct-current installation. There will be no increase in the size of the lighting wires, due to the power factor of .80 taken for the entire ship installation, because, as before noted, the lighting load is practically non-inductive.

Recently the tungsten lamp has come into extensive use commercially, and recent tests of this lamp on board ship have been very satisfactory. This lamp has a much higher efficiency than the carbon-filament lamp now in use and lasts as long. The current consumed by the tungsten lamp is 1.20 watts per candle against 3.5 watts per candle for the carbon-filament lamp.

Assuming the battle load on the lighting circuits of a battleship to be 110 kilowatts, it can be readily seen that if the tungsten lamps were used exclusively, this load of 110 kilowatts could be reduced to nearly one-third of its present value, bringing the lighting load down to about 40 kilowatts. By this method we can reduce the size of the present lighting wiring and consequently the weight and cost in the same ratio.

From a table of weights and costs of wiring as computed for a recent battleship it is found that the weight of copper in the lighting system is approximately 10,000 pounds, mains, feeders and bus bars, costing about $4,500.00. Computing the same for the exclusive use of tungsten lamps a very conservative estimate places the weight of copper at 6,000 pounds and the cost at $3,500.00, saving in weight of 4,800 pounds and a cost of $1,000.00.

The lighting system as proposed seems to be more desirable for shipboard use for the reason that the lights will be steadier at all times due to the better voltage regulation of the alternating-current generators when fitted with the Tirrill regulator. Considering the economy of operation and the durability, one system does not appear to possess any advantages over the other when used with the same type of lamp.

POWER WIRING.

In the alternating-current system all power appliances are supplied with 220-volt, three-phase, sixty-cycle current. The voltage regulation remains the same as in the direct-current installation.

The reactive drop ordinarily encountered in three-phase power mains can be assumed to be negligible, due to the fact that all power mains will be run in conduit, and that all circuits are short and direct.

The ratio of the weights of copper in the mains of the two systems, as the length is constant, will depend upon the square of the voltage used. The change from a two-wire direct-current to a three-wire alternating-current system reduces the weight and the area of the wiring in the ratio of four to three; but to allow for the power factor of .80 we must increase the area of the alternating-current wiring in the ratio of 1 to .8 making a final ratio, considering these three factors, between the area of the alternating-current and direct-current wiring of .277. The area of the electrical circuit is twice the area of each conductor in the case of the direct-current, and three times the area in the case of the alternating-current system. Thus the factor for conversion from the area of a given direct-current conductor is $.277 \times 2/3 = .185$.

From the table of power wiring, bus bars, feeders and mains computed for the new battleships it is found that the weight of copper is approximately 25,000 pounds and will cost about $10,000.00. If the proposed alternating-current wiring were substituted the weight will be reduced to about 10,000 pounds and the cost to about $5,000.00, a saving in weight of about 15,000 pounds, which is sixty per cent. of the weight of the present installation, and in cost of about $5,000.00, which is fifty per cent. of the total cost.

INTERIOR COMMUNICATION WIRING.

Circuits at present operated with direct current at a voltage of 20, supplied by motor-generators, will be operated as at present, the power being supplied by induction-motor genera-

tors. Circuits now operated at 125 volts, direct current, power supplied by main generators, we propose to operate with alternating current at a voltage of 125, supplied by a small transformer from the power circuit of the ship. Circuits now operating at 125-volt alternating current will be operated direct from the ship's power mains at a voltage of 220.

When the proposed system reaches a higher state of development, it will be perfectly feasible to operate all interior-communication circuits, excepting the telephone circuit, with alternating current; the motor generators at present installed would be replaced by small transformers. This change would give a saving in weight, cost and space.

The telephone circuits at present operated on a voltage of 20 furnished by motor-generators could be more satisfactorily operated by duplicate storage batteries, using each alternately, which would permit of time for recharging of the batteries. A storage battery is more suitable for telephone work, as it operates without scratching of brushes or humming noises which are ordinarily transmitted to the telephone receivers. The storage batteries could be easily charged direct from the ship's circuit by the use of the mercury arc rectifier.

WIRELESS PLANT.

The wireless installation as at present designed for battleships requires a capacity of from 5 to 10 kilowatts, alternating current, single phase, 500 cycles. The alternating-current system, as proposed for a ship's installation, is of three-phase, 60 cycles, consequently the ship's power mains can not be used as a direct source of supply for wireless purposes. In the direct-current installation a motor generator is employed to supply the power for the wireless. This method of supplying power must be retained in the alternating-current installation with the exception that an induction motor will be used for the motor end of the motor generator.

SEARCHLIGHTS.

All searchlights of the Navy are operated on the power circuit and require about 125 ampères at 60 volts. This voltage is now obtained by the employment of heavy rheostats cutting down the voltage from 125 to about 60 volts. These rheostats must be capable of varying the voltage from 60 to 80 volts, for the satisfactory operation of the lights. A great economical advantage would result if the dead resistances in the circuit could be dispensed with, as nearly one-half of the power supplied to the searchlights is dissipated as heat in the resistances. The only way to eliminate this large loss is by the use of motor-generators of one type or another which will supply the reduced voltage.

If the present type of direct-current searchlight is to be retained in the proposed alternating-current system, motor generators will have to be employed for supplying the direct current at 80 volts. This method of supplying power to the searchlights is employed in the new Argentine battleships now building in this country. Of course, there will be an increased cost in installing motor generators, but at the same time there must be considered the fact of the greater economy of operation combined with better efficiency and service.

The searchlights use 60 to 80 volts and 125 ampères each. The current supplied being at a voltage of about 120, the difference, 40 volts, at 125 ampères, will equal 5 kilowatts per light, making a total dissipation of 80 kilowatts for sixteen lights. This loss is quite large, being about seven per cent. of the total generator capacity of the ships.

Alternating-Current Searchlights.—In this country there has been little or no development of an efficient alternating-current searchlight, but this type of searchlight has not only been developed in Germany but it has been installed in all of the latest battleships of that country. The electrical installation of these ships being of the direct-current type, motor generators have been installed in order to obtain the alternating current for the searchlights.

Thus, the alternating-current searchlight is considered in Germany to be superior to the direct-current light. There being no demand for an alternating-current searchlight in this country, naturally there has been little or no investigation along this line; but to force the development of such a light there is required only an insistent demand for it.

If the alternating-current installation should ever be placed on naval ships it is recommended that alternating-current searchlights be installed as being much more economical than the direct-current type, since proper voltage could be obtained from the power mains by the use of a step-down transformer, thus eliminating the motor generators which would be required for the direct-current searchlights.

ELECTRIC HEATING.

Electric heaters in certain parts of the ship remote from steam supply, electric cooking ranges and bake ovens, can be operated on alternating current with as good as if not better efficiency than on direct current.

The power required for electric heating and cooking will range between 375 and 400 kilowatts.

MOTORS ABOARD SHIP.

A selection of one of the types of induction motor, previously described, will now be made to fulfil the operating requirements of each motor unit of a battleship installation. The induction motors proposed are such as will meet the present requirements of the specifications for induction motors for navy yard service. The heating requirements, efficiency, rating, etc., are similar to the requirements for direct-current machines now installed.

TURRET-TURNING MOTORS.

The method of control best adapted to fulfil the requirements for these motors is a special mechanical gear manufactured by the Waterbury Tool Company, and which is operated

by a continuously running constant-speed motor. The ordinary induction motor with a low-resistance squirrel-cage rotor is admirably adapted for this purpose. The direct-current specifications call for a motor totally enclosed for this use on account of the danger of sparks from the commutator. With the induction motor all danger of sparks is eliminated, and when the location of the motors is considered, it may be seen that they may be left entirely open in perfect safety.

The proposed alternating-current motors are of 35 horsepower, low-resistance squirrel-cage rotor, induction type, wound for 220 volts, 600 revolutions per minute, and of 90 per cent. efficiency and power factor .90 at normal rated load, slip 5 per cent. The saving in weight in these motors for a battleship would amount to about 9,000 pounds.

BOAT-CRANE MOTORS.

Requirements.—There are two boat cranes, of a capacity of about twenty tons each, specified for each battleship. At full load each crane must hoist its load at a maximum speed of twenty feet per minute, and upon removal of the load the empty hook must be hoisted or lowered at two and one-half times the full-load hoisting speed. The maximum lowering of the working load must not be more than twice that of the hoisting speed under load. The motors, two in number, one for hoisting and one for rotation, must be totally enclosed and watertight, and be capable of standing an overload of fifty per cent. for five minutes, and one hundred per cent. for one minute. The crane must be capable of revolving at the rate of one complete revolution per minute. Both motors to be provided with separate mechanical and electrical brakes. The direct-current motors now installed are of forty horsepower for hoisting and of thirty horsepower for rotation.

The selection of a suitable alternating-current motor for hoist duties presents complications, but there are two types of induction motor which will fulfil the requirements satisfactorily, namely: a polar-wound rotor type of induction motor,

or a squirrel-cage rotor type of induction motor with pole-changing connections to the stator giving the different speeds required for operating the crane hook loaded and unloaded. The alternating-current motor proposed will be of the pole-changing type; this type will give the greatest efficiency at the different speeds.

The direct-current motors at present installed for hoisting are of forty horsepower, 120 volts, 410 revolutions per minute, totally enclosed, and have an efficiency of about 89 per cent. at normal rated load. The direct-current rotating motors are of thirty horsepower, 120 volts, 500 revolutions per minute, and have an efficiency at normal rated load of about 86 per cent.

The alternating-current motors proposed are as follows: For hoisting, a squirrel-cage-rotor induction motor of forty horsepower, 450 revolutions per minute, 220 volts totally enclosed, with an efficiency of about 88 per cent. and power factor .75 at normal rated load, and slip 10 per cent. For rotation, a squirrel-cage-rotor induction motor of thirty horsepower, 600 revolutions per minute, totally enclosed, 220 volts, efficiency 87 per cent. and power factor .74 at normal rated load, slip 6 per cent.

The total weight—motors, brakes and control apparatus—for each crane will be about the same in each installation.

DECK-WINCH MOTORS.

Requirements.—Motors to be compound wound, reversible, enclosed and watertight, and of at least the horsepower required. Windings of motor fields to be such that with no load on the motors the speed will not exceed the full-load speed by more than one hundred per cent. at the rated voltage. Motors to stand fifty per cent. overload for five minutes and a hundred per cent. for one minute. The direct-current motors installed to meet these requirements are of thirty-five horsepower, 350 revolutions, 120 volts, totally enclosed, and of an efficiency at normal load of 86 per cent.

The alternating-current motors proposed are of thirty-five horsepower, polar-wound rotors, connected through slip rings to external variable resistance, induction type, 600 revolutions per minute, totally enclosed, efficiency at full load 86 per cent. power factor .81.

The weights of both installations will be about the same.

The alternating-current installation will give equally as good results as the present direct-current installation. The increase in speed of the alternating-current motors is taken care of by changing the present ratio of the gearing of the winches, the low speed of the present direct-current motors not being suitable for the alternating-current motors.

MOTORS FOR TURNING OVER THE MAIN ENGINES OR TURBINES.

Requirements.—Motors to be compound wound, reversible, totally enclosed, with a speed adjustment of not less than two to one. Motors to be of not less than the required horsepower, and must stand an overload of fifty per cent. for five minutes and a hundred per cent. for one minute. The direct-current motors now in use to meet the above requirements, for turbine battleships, are of five horsepower, totally enclosed, compound wound, 500 revolutions per minute, and of an efficiency of about 75 per cent.

The proposed alternating-current installation consists of five-horsepower motors, squirrel-cage rotors, induction type, totally enclosed, 220 volts, and to have connections for changing the number of poles from 16 to 8 in order to obtain the desired speed change; efficiency about 83 per cent. and power factor .75 at normal load, slip 10 per cent.

The weight of the apparatus would be about the same in both cases.

TURRET-HOIST MOTORS.

Requirements.—Motors to be wound in accordance with the system of control used, to be totally enclosed and watertight reversible and of a speed and horsepower required; and when required, to be fitted with an automatic brake.

The alternating-current motors proposed are of ten and forty horsepower respectively, squirrel-cage rotor, induction type, 220 volts, and of an efficiency of 85 per cent. for the lower and 90 per cent. for the upper hoists, and power factor .85 and .89 at normal load; slip 6 and 9 per cent. There being no danger of sparks from the induction motor they will be of the open type. The saving in weight of each set of motors and control apparatus will be about 550 pounds; and as there are ten hoists in a ship the total saving in weight will be 5,500 pounds.

AIR-COMPRESSOR MOTORS.

Requirements.—The specifications call for direct-connected motors to a four-stage air compressor with a continuous running speed of 375 revolutions per minute.

The present direct-current installation is equipped with a fifty-five horsepower, compound-wound motor, semi-enclosed, 120 volts, 340 to 410 revolutions per minute, and of an efficiency of 89 per cent. at normal rated load. The proposed alternating-current motor to replace the above is an open motor of fifty-five horsepower, 220 volts, squirrel-cage rotor, induction type, of 400 revolutions per minute, with variable resistance or auto-transformer in circuit with the primary which will give the slight variations of speed required. The efficiency will be about 90 per cent. and the power factor .91 at normal rated load, slip 6 per cent.

The saving in weight in favor of the alternating-current apparatus will be about 600 pounds.

GUN-ELEVATING MOTORS.

The ordinary squirrel-cage rotor, induction-type motor, is best suited to meet the requirements for gun elevating. The direct-current specifications call for totally enclosed motors, but open alternating-current motors will be substituted, as there is no danger from sparks with the induction motor.

The alternating-current motors proposed are of fifteen horsepower, 600 revolutions per minute, 220 volts, open mo-

tors, efficiency 87 per cent., power factor .89 at normal full load. The total saving in weight for motors and control apparatus in favor of the alternating-current installation is about 4,000 pounds.

FORCED-DRAFT BLOWERS.

Requirements.—Each motor shall be totally enclosed, shunt wound, and direct connected to the fan. The speed must be capable of reduction by field regulation in at least equal decremants down to twenty per cent. below maximum rated speed.

The direct-current motors at present installed are of thirty-five horsepower, 120 volts, totally enclosed, 610 to 770 revolutions per minute, and have an efficiency at normal rated load of 88 per cent. The alternating current motors proposed are of thirty-five horsepower, polar-wound rotor, induction type. The rotor is connected in circuit with variable external resistance in order to obtain the required variation of speed. Motors to be wound for 220 volts, 770 revolutions per minute, totally enclosed, with an efficiency of 87 per cent., and power factor .82 at normal rated load. The weight of each installation will be about the same.

In place of the above alternating-current motors we could use a motor with squirrel-cage rotor with variable resistance or an auto-transformer in circuit with the primary, which would give variable applied voltage to the stator and, consequently, the small variations of speed required. The second suggestion is more economical than the first as regards power consumption at less than normal load.

RAMMER MOTORS.

The alternating-current motors proposed will be a ten-horsepower, polar-wound rotor, induction motor. The rotor circuit is connected to external resistance through slip rings in order to obtain the necessary speed regulation. Motors to be wound for 220 volts, 900 revolutions per minute, totally

enclosed, efficiency 81 per cent., power factor, 72 at normal rated load.

The weights of both installations will be approximately equal.

FRESH-WATER AND SANITARY-PUMP MOTORS.

Requirements.—Each motor is to be compound wound, open or semi-enclosed, depending upon the location of the motor, and of at least the horsepower required. Motors to stand fifty per cent. overload for one hour without injury to the windings. No speed regulation is required.

We can readily substitute for the direct-current motor at present in use a constant-speed squirrel-cage rotor, induction motor, which is admirably adapted for this duty. It may be left entirely open, and if installed in a damp place the windings may be made moist proof at a slight increase in cost. Motors for this purpose are from three to ten horsepower. The efficiency of the ordinary direct-current motors employed under this head vary from 74 to 80 per cent. The proposed alternating-current motors, if of the open type, will be more efficient, efficiency varying between 80 and 85 per cent.

LAUNDRY AND GALLEY MOTORS.

The requirements of the motors under this head are similar to those mentioned above, except that they are shunt wound instead of compound wound. Being constant-speed motors, all of the remarks made above apply equally here.

FIVE-INCH AMMUNITION-HOIST MOTORS.

The alternating-current motor proposed is a three-horsepower, totally enclosed, with polar-wound rotor connected through slip rings to external variable resistance.

A change of two or three speeds is considered necessary, consequently we propose to use a squirrel-cage rotor, induction-type motor, with connections for changing the number of poles, thereby obtaining the necessary variations in speed.

This motor will be of three horsepower, 600 revolutions per minute, 220 volts, and of an efficiency of 83 per cent., and power factor .86 at normal rated load, slip 8 per cent. The alternating-current motor in the last case to be of the open type, there being no commutator or brushes the necessity of guarding against sparking is eliminated.

The totally-enclosed alternating-current motor is less efficient than the open one, and the power factor is correspondingly low at light loads.

The decrease in weight in favor of the alternating-current motor would be approximately 5,000 pounds. The electric brakes will be of the usual solenoid type.

AMMUNITION CONVEYORS.

The requirements for the ammunition conveyors are similar to those of the five-inch hoists, excepting that no brake is required.

The alternating-current motor proposed is a four-horsepower, squirrel-cage rotor induction type, 600 revolutions per minute, 220 volts, with pole-changing connections for varying the speed. Efficiency of 82 per cent., power factor .83, slip 9 per cent.

VENTILATING-SET MOTORS.

Requirements.—Motors to be shunt wound, direct-connected to the fan, and to have a small speed variation by means of field control, to be enclosed or semi-enclosed, depending upon the location.

The direct-current sets range in horsepower from one and one-half to eight horsepower, depending upon the size of the fan. The following is an example of one set: Motors three horsepower, open, 120 volts, 615 to 810 revolutions, efficiency at normal rated load 83 per cent.

As a substitute it is proposed to install an alternating-current open motor of three horsepower, 220 volts, with pole-

changing connections for two speeds; efficiency at normal rated load 86 per cent., power factor .77, slip 5 per cent. All motors to have squirrel-cage rotors of low resistance. The motors of all other sets would have characteristics similar to these.

The total saving in weight in favor of the alternating-current installation for ventilating-blower motors would be about 10,000 pounds.

POWER-DOOR AND HATCH MOTORS.

Requirements.—Motors to be shunt or compound wound, reversible, and totally enclosed. The horsepower of the motor depending upon the weight and type of the door or hatch. There are two speeds required, one of about 600 revolutions per minute for closing, and the other of about 800 revolutions per minute for opening.

The alternating-current motors with squirrel-cage rotor, induction type, with connections for pole changing for two speeds, will fulfil all requirements. The rotor to be of high resistance which will produce a slip of about ten per cent, thereby giving sufficiently large starting torque.

MOTOR GENERATORS.

The motor generator being a constant-speed machine we can immediately adapt the squirrel-cage-rotor type of induction motor with slow-resistance secondary. This type of induction motor is inherently a constant-speed machine, and is most admirably adapted for the motor ends of motor generators. By the employment of the induction motor all motor generators will be reduced in weight by about twenty per cent.

CAPSTAN MOTORS.

Requirements.—Motors to be compound wound, reversible, open type, and of the speed and at least the horsepower output required. There are five different speeds required in each direction, and the motor is to be fitted with an efficient elec-

tric brake. The present direct-current motor installed is of forty horsepower, 120 volts, 550 revolutions per minute, and of an efficiency at normal rated load of 89 per cent. The proposed alternating-current motor for this duty will be of the polar-wound rotor, induction type, 600 revolutions per minute, 220 volts, open type, efficiency 90 per cent. and power factor .89 at normal rated load. The motor will be geared to the capstan in accordance with present practice.

There are several other applications of motors for power purposes aboard ship, such as portable ventilating sets, machine-shop tools, etc., all of which can be successfully operated by alternating-current motors.

From a consideration of the foregoing description of motors and their duties, it would appear that the induction motor in some one of its forms will fulfil the various requirements now demanded of the direct-current motor. In all cases where a constant speed is used the induction motor will be found to be superior to any direct-current motor, the speed of the induction motor being practically constant under varying load.

The constant-speed motor can be used for the following purposes aboard ship:

All motor generators.
Turret-gun elevating motors.
Turret-turning motors.
Turret-hoist motors.
Laundry motors.
Galley motors.
Sanitary-pump motors.
Fresh-water pump motors.

In cases where wide variations in speed, not more than four in number, are required, the multi-speed type of the induction motor (speeds obtained by changing the number of motor poles) is superior to the direct-current motors, because at each speed the motor is running near its synchronous speed and has the characteristics of a motor of similar synchronous winding.

This type of induction motor can be used in the following cases:

Ventilating-blower motors.
Turbine-jacking motors.
Crane-hoisting motors.
Crane-rotating motors.
Five-inch ammunition-hoist motors.
Ammunition-conveyor motors.
Long-arm system motors.
Machine-tool motors.

In case many changes of speed are required, the induction motor with polar-wound rotor, or the induction motor with a variable resistance, or auto-transformer in circuit with the primary, may be used; the former method is preferable where the motor has to start under load and the latter where small starting torque is required. These forms of motor are only applied to more or less intermittent-duty motors, as the following list will show:

Crane-hoist motors.
Deck-winch motors.
Rammer motors.
Air-compressor motors.
Capstan motors.
Forced-draft blowers.

Where any of the motors in the preceding list do not require large starting torque, such as forced-draft blowers, the slight variations of speed required can be obtained by the second method described, the motors in this case having rotors of the squirrel-cage type. This method gives fairly good speed control but at the expense of starting torque.

The great advantage of the alternating-current motor lies in its reliability. The absence of a commutator makes the machine very much more simple, requiring less care in operation, with attendant less repairs, which, naturally, will add to the length of service to be rendered by the machine. The average weight of the alternating-current induction motor is from twenty to twenty-five per cent. less than the weight of the direct-current motor; this is quite a factor in its favor when the question of weight in modern ship construction is receiving such prominent attention from shipbuilders. The difference in cost between induction motor and the present direct-current motor will be in favor of the alternating-cur-

rent motor, but this fact will have little influence, as this difference will be small when compared with the total cost of a ship.

SUMMARY.

In concluding it may be said that the alternating-current system on a battleship presents the following advantages over the present direct-current system:

1. Greater reliability.
2. Decreased weight, approximately 25 per cent. in the case of a battleship.
3. Increased economy of operation of generators,—about 23 per cent.
4. Higher efficiency of generators and motors,—3 to 5 per cent.
5. Decreased first cost,—about 20 per cent.
6. Decreased cost of maintenance.
7. Better voltage regulation.
8. Generators capable of handling overloads with greater safety.
9. Longer life of generators and power units.
10. Less attendance required.
11. Absence of electrolysis.

THE WORK OF THE U. S. NAVAL RADIO-TELEGRAPHIC LABORATORY.

By L. W. Austin, Ph. D., Associate.

In the spring of 1908 a suggestion was made to Commander Davis and Lieut. Commander Sweet, then in charge of the wireless work under the Bureau of Equipment, that it would be desirable for the Navy Department to have a laboratory for the investigation of various problems in wireless telegraphy. As soon as it was learned that such a laboratory was contemplated, Professor S. W. Stratton, Director of the Bureau of Standards, kindly offered to place at the disposal of the Navy Department quarters for the new enterprise and to aid it with a portion of the required laboratory apparatus and the use of some of the general equipment of his Bureau. Arrangements for beginning the new work were completed in the course of the summer and work was actually begun early in the autumn. While the Bureau of Standards has been the headquarters of the laboratory, much of the work, from its nature, has had to be done outside, in wireless stations under the control of the Department and on the ships of the Navy.

The problems presented to the laboratory extended over a wide field, as up to this time the art of wireless telegraphy had in America been largely developed without any very exact quantitative knowledge of the factors involved. Wave length, capacity and inductance were the only quantities ordinarily measured. The high-frequency currents, above the range of the ordinary 6-ampère hot-wire meters, were usually measured by means of unknown shunts which gave practically no clue to their actual values. As for the energy losses in the inductances, condensers, spark gaps, etc., no quantitative measurements were for the most part even attempted. In receiv-

ing, a few measurements had been made of the antenna current at short range, while for the greater distances the relative intensity of the received energy was measured by the Fessenden Company by means of the shunted telephone method, a method which up to very recent times has apparently been viewed with great distrust by other workers. From this it will be seen that no quantitative knowledge of the efficiency of sending sets was possible nor was there any well defined conception of the laws relating the energy sent out from the sending antenna and that received at the receiving antenna, beyond a distance of a few miles.

In Europe things were slightly better, the energy losses in high-frequency circuits were in some cases measured with more or less accuracy by means of observations on the logarithmic decrement of the circuits, while Duddell and Taylor, in England, and Tissot, in France, had made actual measurements of the current in receiving antennas up to a distance of about 50 miles.

In what follows I intend to review briefly the general nature of the work which has been carried on during the last three years in connection with the laboratory, and then give brief abstracts of some of the more important publications which have been issued by the laboratory.

During the first year there was but one assistant for the work, Mr. Clark, the sub-inspector of wireless stations, who gave as much of his time as could be spared from his other duties. Since that time a number of chief electricians have been detailed as assistants at different times for longer or shorter periods, while some others have been sent to the laboratory for instruction for special duty in connection with wireless. With the exception of Mr. Clark, none of these men had technical training fitting them to be laboratory assistants, so that it was necessary in each instance to give much instruction before they were capable of being of use. If it had been considered feasible to have obtained civilian assistants with university education the work would undoubtedly have gone much more rapidly. As it is, however, some of the enlisted

men have shown great aptitude for the work and under direction have proved efficient assistants.

The first work undertaken, September, 1908, was the investigation of some Poulsen wireless telephone sets recently purchased by the Government, one of which was installed at the Washington Navy Yard. The electric waves in this set were produced by an arc formed in alcohol vapor, the current being supplied from a 500-volt direct-current generator. Fair results were obtained, conversation being distinctly heard at times over distances up to 40 miles. The conclusions reached, however, were that the set required more skilled attention than would be easily available on shipboard, and that, with its separate motor generator and somewhat bulky apparatus, it would take up too much space to be used as an auxiliary in the ordinary wireless room.

After the conclusion of this test work was taken up at the laboratory at the Bureau of Standards. The most important problems which presented themselves at the beginning had to do with the development of methods of exact measurement of the various quantities involved in wireless telegraphy. A few of these we were able to borrow from European practice, while many others we have had to develop for ourselves.

Among the methods of measurement which had to be devised during the first year of the work were those for the determination of the current in the receiving antenna in the case of signals too weak to be detected quantitatively by the older methods, the determination of the sensitiveness of telephones and detectors, the comparative efficiency of receiving sets, and the determination of antenna resistance. These problems were all solved more or less satisfactorily, Mr. Clark assisting in the work and makng many valuable suggestions.

During this year, in addition to the development of methods of measurement, a systematic study was made of the current distribution in coupled receiving circuits under different conditions of damping of the incoming waves. This work led to interesting results concerning the energy relations between the two circuits and showed especially the necessity of

a variable coupling between them without which neither the full loudness of the signal could be obtained under varying conditions, nor was it possible to get satisfactory sharpness of tuning.

Early in 1909, a contract had been given to the National Electric Signaling Co. to equip with apparatus the 100-kw. high-power station about to be erected in the neighborhood of Washington (Arlington Station). This apparatus was installed at the large station belonging to the company at Brant Rock, Mass., and during the late summer and autumn of 1909 preliminary measurements on the constants and efficiency of the set were carried on at that place. Some determinations were also made of the intensity of the signals sent out by this apparatus at the wireless station of the Washington Navy Yard. It was hoped that in this way some estimate could be made of the range of the new high-power station.

In December of 1909 and the early spring of the following year, experiments were carried on between the Brant Rock station and the scout cruisers *Birmingham* and *Salem*, which were fitted with Fessenden apparatus, for a further test of the 100-kw. set. During this time, at the laboratory a systematic study of the high-frequency resistance of antenna wire, sending condensers, and sending and receiving inductances, was taken up. This, with the testing of various special pieces of apparatus, occupied the time until July 1, 1910, when the most important piece of work thus far engaged in by the laboratory was begun. It had been arranged that the final test between the sets of the scout cruisers and between the cruisers and Brant Rock should take place at this time, and this gave us an opportunity to make a quantitative study of the relation between sending and receiving antenna currents up to a distance of about one thousand miles. The results of these observations more than fulfilled our hopes and showed clearly that, during the day time, at least, perfectly definite laws control the strength of the received signals.

After the close of these tests it was thought desirable to verify the laws already discovered, for smaller vessels and shorter

wave lengths. For this work the torpedo boats *Bailey* and *Stringham* were detailed, and the observations amply verified the general truth of the laws discovered in the Brant Rock test. These and succeeding tests with the same vessels showed also how much can be done even with mediocre apparatus when properly handled. With old type 120-cycle, 2-kw. sets and with masts having an average height above the water of only 37 feet, signals were successfully exchanged at a distance of 155 miles. Raising the antennas to a height of 65 feet extended this distance to 250 miles. It is confidently expected that with their new 500-cycle sets giving 14 or 15 ampères in the antenna, 350 miles will be reached.

During the past year work on the losses in various parts of sending and receiving circuits has been continued, especially interesting results having been obtained regarding the high-frequency resistance of antennas at various wave lengths. In addition to this many comparisons have been made of the efficiency of different types of spark circuits and a study is being carried on of the value of musical quality in the sparks of the sending station.

The quantitative results obtained in the various lines of work have in many cases led to interesting theoretical conclusions, and these have in general shown that the sending and receiving circuits used in wireless telegraphy conform in their action to the mathematical theories given by Hertz and his successors, and, with the exception of the absorption of the waves in passing from one station to another, involve no new or untried mathematical principles such as have often been conceived by the practical workers in radio-telegraphy.

It is hoped that in the future the work of the laboratory will be greatly enlarged, a portion of it being still carried on at the Bureau of Standards and a portion at the Arlington station; while, as soon as more high-power stations are erected, extensive experiments must be carried out on the transmission of signals over greater ranges than have hitherto been studied. Having given a general account of the activities of the labor-

atory, it will perhaps be of interest to consider particularly some of the more important work which has been done.

MEASUREMENT OF ELECTRICAL OSCILLATIONS IN THE RECEIVING ANTENNA.

From the very beginning it was felt that the progress of radio-telegraphy was greatly hampered by the lack of the means for quantitative measurements of the energy at the receiving station. Various methods were experimented with, and finally the following arrangement was adopted as most satisfactory and accurate. The connections are shown in Fig. 1. A, E is the receiving antenna, L_3, L_4, K, D is an ordinary

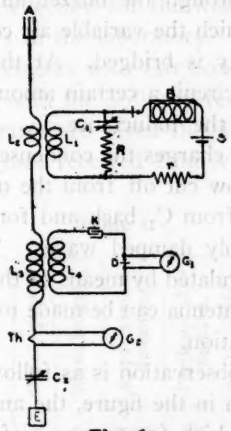


Fig. 1.

coupled receiver with variable coupling between the primary and secondary. D is a perikon detector to which is attached a sensitive galvanometer G_1 instead of the telephones. Special experiments showed that the deflections of the galvanometer attached to the detector are proportional to the oscillatory energy passing through the detector, and also proportional to the square of the current in the antenna A, E. In order to calibrate the deflections of the galvanometer G_1 in terms of the oscillatory current in the antenna, it was necessary to in-

sert a very sensitive thermoelement, Th, in the antenna ground. When the high-frequency currents pass through the thermoelement it becomes heated and produces a current in the galvanometer G_2. This heating is proportional to the square of the antenna current, therefore the deflection in G_1 will be proportional to that in G_2, so if the thermoelement is already calibrated so that its galvanometer deflections can be reduced to microampères, we have at once a method of calibrating the detector D with its galvanometer. For the artificial excitation of the antenna a tuned buzzer circuit L_1 C_1 was added to the apparatus. The action of this buzzer circuit is exceedingly interesting; the current from the storage battery S flows through the buzzer and through the inductance L_1 across which the variable air condenser, C_1 of 0.005 microfarad capacity is bridged. At the moment before the buzzer breaks the circuit a certain amount of electromagnetic energy resides in the inductance L_1. When the circuit is broken this energy charges the condenser, and as this portion of the circuit is now cut off from the other, the charge continues to oscillate from C_1 back and forth through L_1 in the form of very feebly damped waves. The exact degree of damping can be regulated by means of the resistance R so that the action on the antenna can be made to simulate any desired type of sending station.

The method of observation is as follows: With the circuits connected as shown in the figure, the antenna is tuned to the incoming signals, which for purposes of measurement should preferably be long dashes, and the coupling between L_3 and L_4 adjusted so that the galvanometer attached to the perikon gives a maximum deflection. When the signals have ceased, the buzzer-exciter circuit L_1 C_1 is adjusted to resonance with the antenna circuit and coupled to it just closely enough so that a readable deflection is produced on the thermoelement galvanometer, while a large deflection is produced on the galvanometer attached to the perikon. The ratio of these deflections enables us at once to calculate the number of microampères flowing in the antenna corresponding to any given

deflection of the galvanometer, assuming that the deflections are proportional to the square of the current. Still greater accuracy may be obtained by adjusting the exciter circuit used in the comparison of the thermoelement and perikon to the same decrement as that of the sending station. If the effective resistance of the receiving circuit as a whole be known, the amount of the received energy can be calculated, thus giving an opportunity for experimental determination of the ratio of radiated and received energy under different circumstances and at different distances.

The galvanometer used with the perikon is considerably more sensitive than any telephone, for long dashes it gives about 3 millimeters deflection for the weakest signals audible in the best telephones in the laboratory. This makes it possible to make measurements with fair accuracy on any signals which are detectable with the ordinary receiving apparatus.

In the following table are given the data of a number of measurements of signals received from various stations.

Table I.

Sending station.	Rated power.	Wave lengths.	Approximate distance.	Perikon deflection.	Ratio perikon to thermo-element.	Received current.	Effective receiving resistance.	Received energy.
	kw.	m.	miles.	mm.		10^{-6} amp.	ohms.	10^{-6} watts.
Philadelphia (Navy Yard)	3	750	100	32	457	56.3	25	8
Brant Rock	60	3,750	350	330	1,280	108	80	93.5
U. S. S. *Salem* (at New York)	10	1,000	185	100	540	91	38	31.5
U. S. S. *Birmingham* (at sea)	10	1,000	300	115	540	98.3	38	36.5
Washington (Navy Yard)	2	900	6	Read directly on thermo-element.		1,400	25	4,900

1 mm. on thermo-element galvanometer $= 213.10^{-6}$ amp. A. C.

The table is self explanatory. The higher land absorption is well shown in the cases of the *Salem* and the *Birmingham*. With the same power and approximately the same amount of

energy radiated, the *Birmingham* delivered more energy at the receiving station at a distance of 300 miles than the *Salem* at 185, the path of the *Salem's* radiation being all the way overland, while the *Birmingham's* was for more than two-thirds of the way oversea.

The *Birmingham's* radiated energy may be calculated as follows: The height of the flat-top antenna was approximately 40 meters and the wave length 1,000 meters; from the Hertz radiation formula

$$R = 1{,}600\,\frac{h^2}{\lambda^2} = 2.57 \text{ ohms},$$

The antenna current was 33 ampères.

The radiated energy

$$E = 33^2 \times 2.57 = 2.810^3 \text{ Watts.}$$

The ratio of received to radiated energy

$$\eta = \frac{36.5\,.10^{-8}}{2.8\,.10^3} = 1.30\,.10^{-10}$$

For stronger signals the received energy may, of course, be measured directly on the thermoelement in the antenna, the perikon circuit being removed or coupled so loosely that no appreciable energy is taken up.

QUANTITATIVE EXPERIMENTS IN LONG-DISTANCE RADIO-TELEGRAPHY.

As has been already mentioned, the tests of the Fessenden apparatus at Brant Rock and on the scout cruisers *Birmingham* and *Salem* gave an opportunity to investigate experimentally the relation between the currents in sending and receiving antennas at distances greater than had hitherto been attempted.

The Brant Rock station developed 60 kw. Its antenna was an umbrella 420 feet high at top and about 170 feet at bottom, with a capacity of 0.0073 microfarad. The scouts' sets

were of 10 kw. and their antennas were flat-topped, 130 feet high, with a capacity of 0.0018 microfarad. During a part of the test this was increased to 0.0025 by extensions slanting downwards fore and aft of the regular antenna. The chief wave lengths used were 1,000 meters and 3,750 meters on the ships, and 1,500 meters and 3,750 meters at Brant Rock.

In the July, 1910, test, which was considered the most reliable, the measurements were carried on up to nearly 900 nautical miles between the ships and to about 1,200 miles between Brant Rock and the *Birmingham*.

The receiving measurements for the shorter distances were made in part by means of thermoelements in the antenna. At the greater distances the method of telephone shunt readings in connection with the electrolytic detector was used, the value of the received antenna current being derived from the shunt reading by means of comparison readings taken on a thermoelement in the antenna at the shorter distances.

On account of heavy atmospheric disturbances, but few measurements could be taken with the perikon detector and galvanometer.

Table II.—"BIRMINGHAM" RECEIVED AT BRANT ROCK, JULY.

Nautical miles	d Km	I_s=33 amp. λ=1000 m. K=475000 10^{-6} amp.			I_s=27 amp. λ=3750 m. K=138000 10^{-6} amp.		
		$\frac{K}{d}$ 10^{-6} amp.	I_R 10^{-6} amp.		$\frac{K}{d}$ 10^{-6} amp.	I_R 10^{-6} amp.	
			Obs.	Cal.		Obs.	Cal.
20	37	12,830	11,500	12,200	3,730	3,500	3,620
50	92.5	5,135	4,600	4,460	1,492	1,400	1,385
100	185	2,565	2,100	1,940	746	700	645
200	370	1,283	800	737	373	260	280
300	555	856	260	370	249	120	161
400	740	642	130	212	187	60	105
500	925	514	80	128	149	40	73
600	1,110	428	50	81.5	124	30	52
800	1,480	321	35	35	93	24	29
1,000	1,850	257	20	15.4	75	22	18

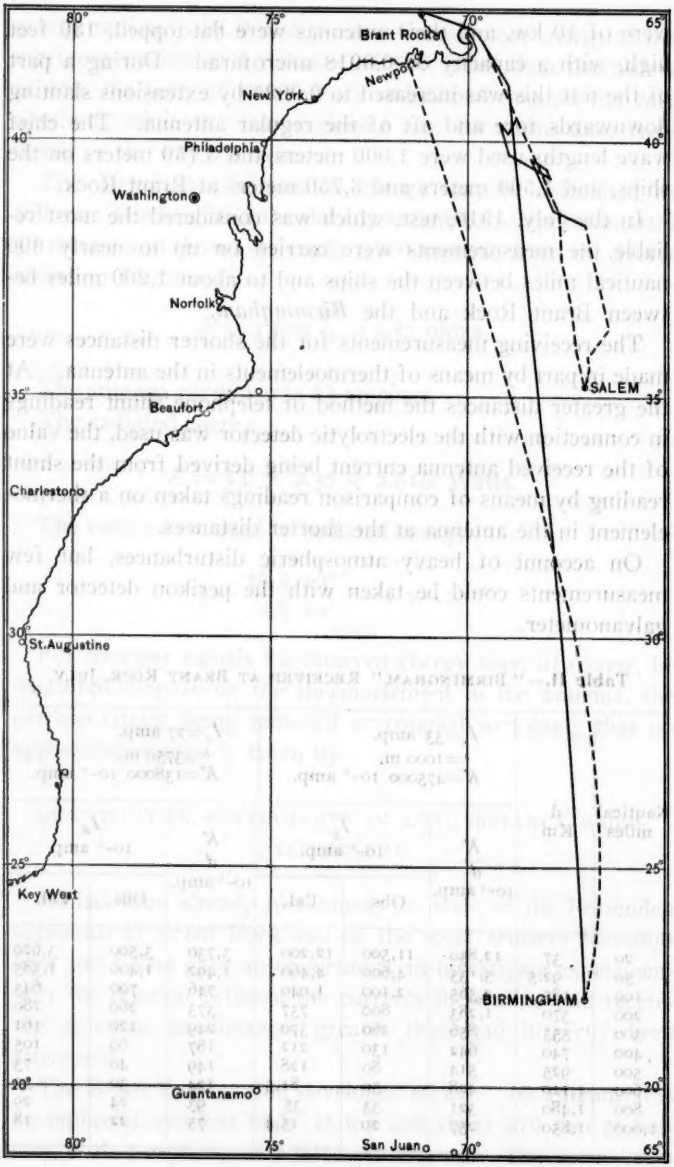

80°
75°
70°
65°
Brant Rock
Newport
New York
40°
Philadelphia
Washington
Norfolk
35°
Beaufort
SALEM
Charlestono
30°
St. Augustine
25°
Key West
BIRMINGHAM
20°
Guantanamo
San Juano

Fig. 2.

The course of the ships during the July test is shown on the map of Fig. 2. As an example of the observations made Table 2 is given, showing the received currents at Brant Rock when the *Birmingham* was sending from various distances at the two wave lengths 1,000 meters and 3,750 meters. The observed values in the table were taken from the smoothed curve of observation, while the calculated values were taken from the equation

$$I_R = \frac{K}{d} e^{-\frac{\alpha d}{\sqrt{\lambda}}}. \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

I_R represents the received antenna current, λ the wave length, and d the distance in kilometers. K and α are constants while e is the base of the natural logarithms. This equation shows that the electrical waves decrease in intensity in proportion to the distance, as found by Duddell and Taylor. In addition they are subject to an absorption which varies with the wave length and which may be expressed mathematically by the term $e^{-\frac{\alpha d}{\sqrt{\lambda}}}$.

This is true in general for day transmission. The absorption at night was found to be entirely irregular, varying from zero to the day value, but is less during the winter than in summer. The great variations in night absorption make useless all attempts to judge the quality of wireless apparatus from night distances. The extraordinary distances of wireless communication sometimes mentioned in the daily papers are probably due to a reflection of the waves from the conducting layers of the upper atmosphere. The conditions which produce this reflection are too erratic and infrequent to be counted on for purposes of communication. It was observed during the experiments that variations occur also during the day time, but these, except in rare instances, are small.

In Tables III and IV the records of the signals exchanged between the *Salem* and *Birmingham* are given, while Fig. 3 represents graphically the received antenna currents at Brant Rock when the *Birmingham* was sending at a wave length of 1,000 meters. This shows the irregularity of the night sig-

nals and also indicates the large value of the absorption at the greater distances, the dotted line indicating the current that should have been received if the absorption had been zero.

In addition to the measurements already mentioned, subsidiary observations were made between Brant Rock and a small station eleven miles away on the effect of the height of a flat-top antenna on sending and receiving, and also on the effect of wave length. It was found that the received an-

Table III.—"SALEM" RECEIVED ON "BIRMINGHAM," JULY.

I_s=27 amp. λ=1060 m. K=230000 10^{-6} amp.

Nautical miles.	d Km	$\frac{K}{d}$ 10^{-6} amp.	I_R 10^{-6} amp. Obs.	I_R 10^{-6} amp. Cal.
20	37	6,220	.....	5,880
50	92.5	2,486	.....	2,160
100	185	1,243	.....	941
200	370	622	325	357
300	555	414	165	179
400	740	311	95	102
500	925	249	65	62
600	1,110	207	45	39.4
800	1,480	155	19	17.1
1,000	1,850	124	10	7.5

Table IV.—"BIRMINGHAM" RECEIVED ON "SALEM," JULY.

I_s=33 amp. λ=1000 m. K=185000 10^{-6} amp.

Nautical miles.	d Km	$\frac{K}{d}$ 10^{-6} amp.	I_R 10^{-6} amp. Obs.	I_R 10^{-6} amp. Cal.
20	37	5,000	.....	4,730
50	92.5	2,000	.....	1,740
100	185	1,000	.....	757
200	370	500	260	287
300	555	333	130	144
400	740	250	75	82.4
500	925	200	50	49.9
600	1,110	167	35	31.7
800	1,480	125	16	13.8
1,000	1,850	100	10	6.2

tenna currents are proportional to the product of the heights of the sending and receiving antennas, and inversely proportional to the wave length, provided the antenna resistances remain constant. How important the effect of antenna height is may be judged when it is remembered that the strength of the received signal varies with the square of the received antenna current. It will therefore be proportional to the square of the product of the sending and receiving antenna heights.

Taking account of the influence of antenna height and wave length, equation (1) may be extended and a general day transmission formula written as follows

$$I_R = 4.25 \frac{I_s h_1 h_2}{\lambda d} e^{-\frac{\alpha d}{\sqrt{\lambda}}} \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where h_1 h_2 are the antenna heights, *Is* the sending antenna current, and the other symbols have the significations already given. The currents are given in ampères, and all lengths in kilometers. From this it would appear that it is advisable to rate stations according to the magnitude of the antenna current, or, perhaps better, according to the product of the current into the height.

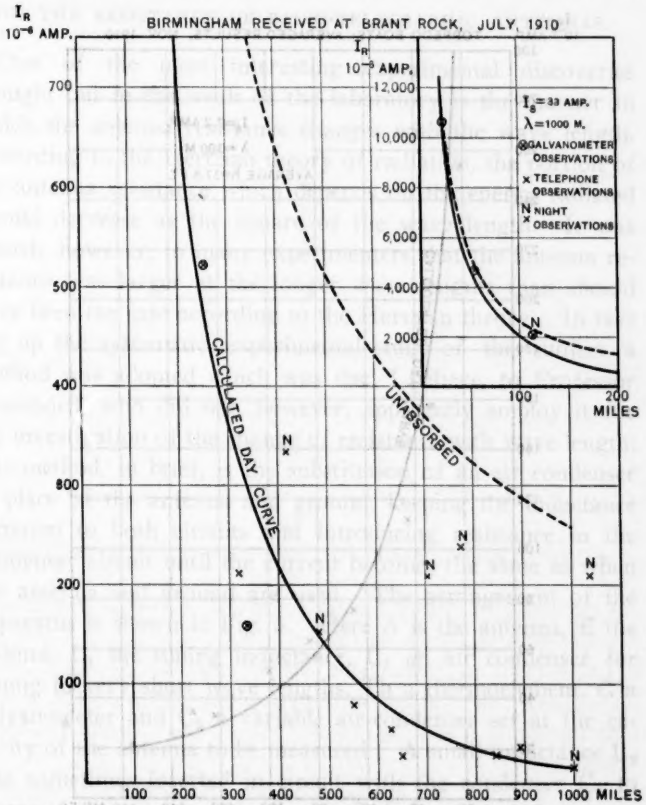


Fig. 3.

As has been already mentioned, after the close of the Brant Rock tests, experiments were made on the torpedo boats *Bailey* and *Stringham*, in order that the accuracy of the formula might be tested for small antennas and short wave lengths. The curve of Fig. 4 shows how admirably the observed received currents followed the calculated curve.

Another interesting fact worthy of mention was brought out during the Brant Rock test. While the *Birmingham* lay in Newport signals were sent to Brant Rock with wave lengths of 1,000 and 3,750 meters. It is well known that the region

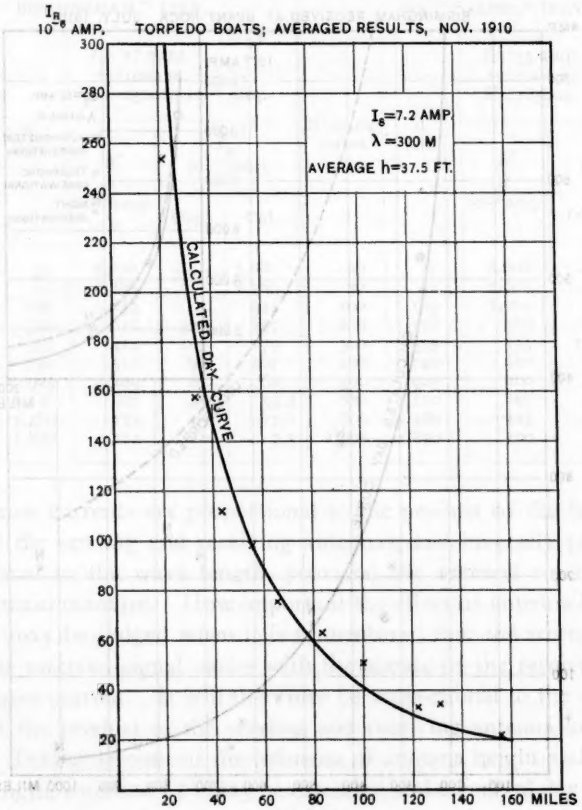


Fig. 4.

north of Newport absorbs electrical waves very powerfully. The experiments showed how much this ground absorption depends on wave length, for while the 1,000-meter signals from the *Birmingham* lost over 90 per cent. of their energy before reaching Brant Rock, the 3,750-meter signals came in without measurable loss, being as strong as though they had passed over salt water. This shows the importance of choosing suitable wave lengths for working between land stations where strong absorption is found.

ON THE RESISTANCE OF RADIO-TELEGRAPHIC ANTENNAS.

One of the most interesting experimental discoveries brought out in the work of the laboratory is the manner in which the antenna resistance changes with the wave length. According to the Hertzian theory of radiation, the portion of the antenna resistance which depends on the energy radiated should decrease as the square of the wave length. It was known, however, to many experimenters that the antenna resistance was larger at the longer wave lengths than should have been the case according to the Hertzian theory. In taking up the systematic experimental study of this subject, a method was adopted which was due, I believe, to Professor Fessenden, who did not, however, apparently employ it for the investigation of the change of resistance with wave length. His method, in brief, is the substitution of an air condenser in place of the antenna and ground, keeping the inductance common to both circuits and introducing resistance in the condenser circuit until the current becomes the same as when the antenna and ground are used. The arrangement of the apparatus is shown in Fig. 5. Here A is the antenna, E the ground, L_1 the tuning inductance, C_1 an air condenser for tuning to very short wave lengths, Th a thermoelement, G a galvanometer and C_2 a variable air-condenser set at the capacity of the antenna to be measured. A small inductance L_2 was sometimes inserted in circuit with the condenser C_2 to represent the antenna inductance. This has little influence on the results and at least for the longer wave lengths may be

perfectly well omitted. S_1 and S_2 are switches for connecting either the antenna and ground or C_2 to the rest of the circuit. R is a resistance introduced in the circuit C_2 to bring down the thermoelement deflection to the same value as that observed when the antenna and ground are in circuit. The high-frequency resistance consists of separate units of fine constantan wire inserted in mercury cups. The measurement circuit

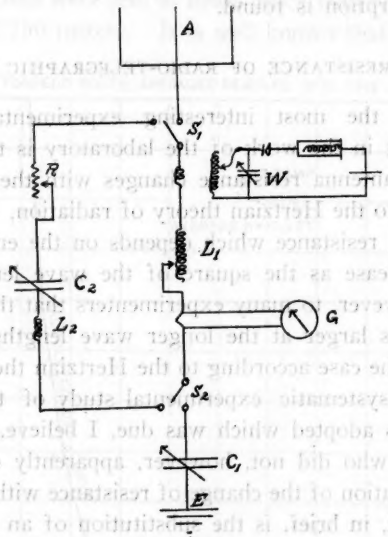


Fig. 5.

DIAGRAM OF APPARATUS FOR MEASURING ANTENNA RESISTANCE.

is excited by a buzzer exciter circuit W of the type previously described and shown on figure 1.

Observations were made on the antenna of the U. S. S. *Dolphin* lying at the Washington Navy Yard, on the antenna of the Navy Yard station and the antenna at the Bureau of Standards.

Fig. 6 shows the curves obtained in these experiments. It is seen that beginning with the short wave lengths the resis-

tance falls rapidly until a point is reached which is not far from twice the fundamental wave length of the antennas. Curve A is the Bureau of Standards, B the U. S. S. *Dolphin*, and C the Washington Navy Yard. In curves B and C, where the ground resistance is low, the resistance rises gradually, while in the curve of the Bureau of Standards, where the ground resistance is larger, the rise of the resistance with

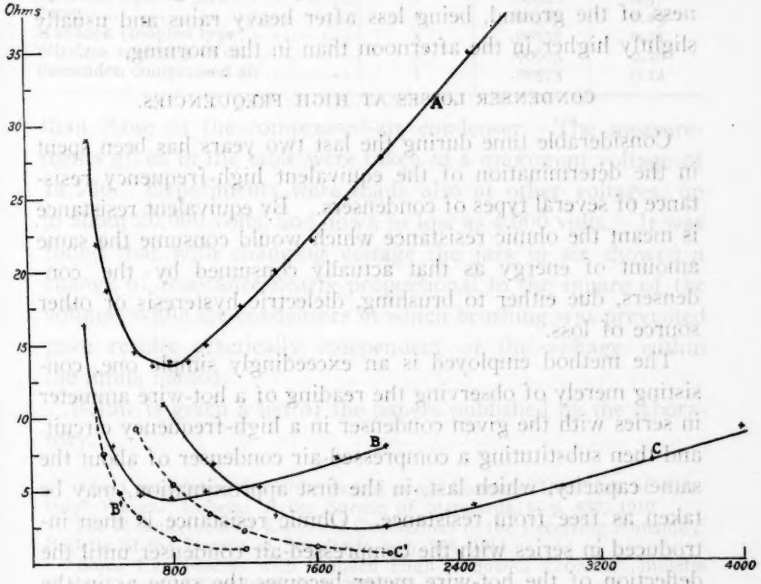


Fig. 6.

DIAGRAM SHOWING THE RELATION BETWEEN WAVE LENGTH AND ANTENNA RESISTANCE.

increasing wave length is more rapid. The theoretical radiation resistance, corresponding to the curves B and C, is shown in the dotted curves B′ and C′. We have here an indication of two factors in the resistance of an antenna, one decreasing approximately as the square of the wave length, while the other increases nearly directly as the wave length. It seems probable that the portion of the resistance which increases as the wave length is ground, or earth current resistance.

Observations have also been made at other stations and on other ships than those already mentioned, but in all cases it was found that the curves followed in general form those of Fig. 6. On some of the ships the minimum resistance fell to about 2 ohms, and in these cases the rise of resistance with increasing wave length was very slight.

In the case of land stations it is found that the resistance and the steepness of the curve vary according to the dryness of the ground, being less after heavy rains and usually slightly higher in the afternoon than in the morning.

CONDENSER LOSSES AT HIGH FREQUENCIES.

Considerable time during the last two years has been spent in the determination of the equivalent high-frequency resistance of several types of condensers. By equivalent resistance is meant the ohmic resistance which would consume the same amount of energy as that actually consumed by the condensers, due either to brushing, dielectric hysteresis or other source of loss.

The method employed is an exceedingly simple one, consisting merely of observing the reading of a hot-wire ammeter in series with the given condenser in a high-frequency circuit, and then substituting a compressed-air condenser of about the same capacity, which last, in the first approximation, may be taken as free from resistance. Ohmic resistance is then introduced in series with the compressed-air condenser until the deflection of the hot-wire meter becomes the same as in the case of the condenser under test. Corrections must then be made for the slight differences in capacity of the condensers compared and for the small resistance of the compressed-air condenser. This last in some experiments fell as low as 0.14 ohm.

Table V shows the results of some of these comparisons. In the table, it is seen, as is to be expected, that the Leyden jars in air show larger resistances than most of the other condensers on account of brushing. The paper condenser also shows considerable resistance due to dielectric losses. The jars in oil have remarkably small losses, only slightly greater

Table V.

Volts=14500 (max.) Current=7—8 amp.
Wave length=1,000 m. (approximately).

Condensers.	Number of units.	Capacity, m. f.	Resistance, ohms.
Wireless specialty (1) }	3	0.00603	1.08
Wireless specialty (2) } Leyden jars	3	.00605	1.19
Telefunken..... } in air.	3	.00612	1.59
United wireless ..... }	2	.00603	1.83
Moscicki (special jar).....	1	.00548	0.57
Paper	1	.00580	2.19
Murdock (molded type).....	3	.00535	0.41
Wireless specialty in oil.....	3	.00605	0.28
Fessenden compressed air	1	.00575	0.14

than those of the compressed-air condenser. The measurements given in the table were taken at a maximum voltage of 14,500. Experiments were made also at other voltages, up to about 20,000 volts, and down as low as 4,000 volts. It was found that with changing voltage the jars in air showed a change of resistance nearly proportional to the square of the voltage, while the condensers in which brushing was prevented gave results practically independent of the voltage within the limits named.

Below is given a list of the papers published by the laboratory.

"The Comparative Sensitiveness of some Common Detectors of Electrical Waves," Bulletin of the Bureau of Standards, 6, p. 527, 1910.

"The Measurement of Electrical Oscillations in the Receiving Antenna[1]," Bulletin of the Bureau of Standards, 7, p. 295, 1911.

"Some Experiments with Coupled High-Frequency Circuits," Bulletin of the Bureau of Standards, 7, p. 301, 1911.

"Some Quantitative Experiments in Long-Distance Radio-Telegraphy[2]," Bulletin of the Bureau of Standards, 7, p. 315, 1911.

"High Spark Frequency in Radio-Telegraphy[1];" "The Slipping Contact Rectifying Detector[1];" "The Resistance of Radio-Telegraphic Antennas[1];" Journal of the Washington Academy, 1, p. 5, 1911.

"Wave Length and Ground Absorption of Electrical Waves;" "Directive Action of the Marconi Bent Antenna at Clifden;" "A Table of Radiation Resistances for Various Wave Lengths and Antenna Heights;" Journal of the Washington Academy, 1, p. 190, 1911.

"Condenser Losses at High Frequencies," Journal of the Washington Academy, 1, p. 143, 1911.

"A Ship's Antenna as a Hertzian Oscillator," Journal of the Washington Academy, 1, p. 275, 1911.

"Thermoelemente fur Versuche mit Hochfrequenzstromen," Physikalische Zeitschrift, 12, p. 1226, 1911.

[1] Also published in the "Physikalische Zeitschrift."
[2] Also published in the "Jahrbuch der drahtlosen Telegraphie."

NOTE ON THE PERFORMANCE OF MARINE TURBINES AT REDUCED SPEED.

By E. Buckingham, Associate.

1. If a certain steam turbine has its best water rate at a given speed of revolution, another turbine of similar stage design, working under the same steam conditions and giving the same power, must be larger in diameter or have more rows of blades in order to have the same water rate at a lower rate of revolution. Under the restrictions of space imposed in battleship design, direct-connected turbines can hardly be made to give their best water rate at much below their full load speed. But since, in our service at least, cruising radius is of prime importance, attention should be concentrated on economy at low speeds, if turbines are to be used at all. It is difficult to draw definite specifications which shall leave the private designer sufficient leeway for the use of his own discretion, but when the Bureau of Steam Engineering designs its own turbines and gives all the time and care needed to the work, better results should be forthcoming than have yet been obtained.

There are two obvious methods of going to work. The simplest is to take the whole available space, design the best turbine possible for full power, and let low speeds take care of themselves when the steam flow is reduced. That is the natural thing to do if the specifications lay the main stress on good economy at high speed, and many designs seems to have been worked out on this primitive plan, which is quite suitable for the simple conditions obtaining in electric-power work. The other way is to set aside a part of the space for cruising stages and take only the remainder for the full-power turbine. The cruising stages are put in at the H.P.

end—sometimes in a separate casing—or are interpolated after the first stage, which is then used at all powers and speeds while a smaller or greater number of the cruising stages is by-passed according to the power required. If the second method of designing is pursued, the economy at full speed will not be so good as if the first plan were adopted, because the full power turbine must be smaller, but the cruising stages can be designed specially for the work they have to do and the total result at low speeds may be better than in the former case.

It may be well to remark, before going farther and to fix our ideas, that the only type of turbine at all worth our considering is the combined wheel-and-drum type; for this has every advantage over the pure multicellular turbine, which is obsolete for naval use, and over the pure drum turbine which is fast becoming so. It is needless to dilate on the various reasons for preferring impulse-wheel stages with partial admission and ample radial clearances at the H.P. end, while the superior merits of the drum construction when large steam volumes are to be handled, as toward the L.P. end, are equally evident to anyone who has noticed the relative number of blade rows per foot in the wheel and drum sections of a combination turbine. The only question at present is how long to make the drum, for it is already settled, both by common-sense theory and by the best practice, that both wheels and drum should be used. If this form of construction is adopted, the natural place for the cruising stages is just after the first stage, for with that arrangement the main first-stage shell is the only one ever exposed to the highest pressure existing in the turbine, and the construction can be lighter than if one of the cruising stages were also fitted for use as a first stage.

Whatever the plan adopted, the majority of the stages must have nozzle or guide-blade areas large enough to pass all the steam needed for full speed of the ship. At lower speeds, these stages will be handling only a small fraction of the steam flow at full power,—perhaps only 1/6 or less at half speed. It is evidently important to know how this great

change in the proportion of steam-flow to area will influence the performance of the stages in question; for unless we are able to predict their behavior at low powers we may make the mistake of preferring a machine which is a little better at full power to one which is a good deal better at low.

Suppose that in making a preliminary sketch design for given power, size, steam conditions, and r.p.m., we settle upon the kind and number of stages to try. It is to be presumed that we have more or less reliable data on the stage efficiency to be expected, for otherwise we are all at sea; and if we have this information, it is a relatively simple matter to determine the curve of expansion through the turbine, on the Mollier diagram. From this curve and the full load water rate, which can be found from it, we determine the areas needed at the terminal pressures of the various stages.

If we now want to find out how the stages already designed will act at low powers, we have first of all to determine the curve of pressure distribution through the turbine. At the start we had the problem; given the total steam flow and the curve of pressure distribution—read off directly from the expansion line on the Mollier diagram—to determine the required areas. Now, however, we are confronted by the more difficult inverse problem; given the steam flow and the areas, to determine the pressure distribution. The steam flow is not, to be sure, given exactly, for it is in fact what we are ultimately trying to find in investigating the behavior of the turbine at reduced speed, so that we can not in any event determine the pressure curve exactly at the first attempt. But a little good guessing as a first approximation will make our first *computed* approximation really a second and sufficiently exact one.

PRESSURE DISTRIBUTION AT REDUCED POWER.

2. This subject of the water rate of turbines under such wide variations of load and speed as occur in marine practice has not usually been treated by writers of books on the steam turbine, most of whom confine themselves to explaining how

to design turbines for fixed conditions and consider the effects only of such small modifications of the conditions as occur in electric power work. But for our purposes, conditions which would be so far from normal as never to occur at all in land practice, become of the first importance, and we have therefore to examine the problem stated above: given the areas, together with the steam flow and pressure distribution for which they were designed, to find the pressure distribution when the steam flow is greatly reduced. There is, it is true, one exceptional case in which this problem is not presented, for if intermediate valves are provided, as has sometimes been done, *e. g.*, on the *Salem,* the areas are not fixed by the construction of the turbine but may be varied at will so as to adjust the pressure distribution to suit the ideas of the operator. But the use of such valves has not in general been found to be of sufficient practical advantage to warrant the extra complication, and this method of regulation may be disregarded for the present, though some further remarks on it will be made later. The two remaining methods of regulation differ greatly in appearance, but not much in the results produced. Throttling by the main valve decreases the steam-chest pressure and therefore the pressure drop through the turbine, while it dries or superheats the steam that reaches the first-stage nozzles. Nozzle regulation, *i. e.*, cutting down the steam flow by closing some of the first-stage nozzles, does not change the pressure drop through the turbine, except in so far as the vacuum may improve, but throws a much greater fraction of the load on the first stage, which is usually the least efficient of all.

The solution of our problem of pressure distribution has doubtless long been familiar to the builders, but so far as I know, it has not appeared explicitly in print till rather recently, and I have seen it given in only one book beside Stodola's monumental treatise. It is probably safe to assume that it will be unfamiliar to some of my readers and therefore worth while giving in spite of the fact that it is no longer new.

3. The most effective way to exhibit the facts will be to il-

lustrate them by means of the following table which represents pressure observations taken on the port turbine of the *North Dakota* during her official trials, reduced from the original data which were kindly placed at the writer's disposal by the Bureau of Steam Engineering. The Curtis turbines of this ship have nine ahead stages with separate valves on the first-stage nozzles. There are no separate cruising stages and no intermediate valves. Eight different sets of conditions tre represented in the table, the speeds ranging from 284 down to 122 r.p.m.

		I	II	III	IV	V	VI	VII	VIII
1	No. of obs.	7	7	6	25	3	6	25	3
2	Nozz's open	20	19.6	13.5	13	10	4	4	3
3	St. Ch. press	260	257	251	218	218	221	233	232
4	1st st'e sh'll	97.1	93	61.3	50	38.3	15.6	15.5	11.8
5	Shell No. 2.	0.672	0.667	0.677	0.682	0.671	0.668	0.735	0.689
6	" " 3.	0.429	0.423	0.411	0.410	0.397	0.502	0.457	0.508
7	" " 4.	0.257	0.258	0.248	0.294	0.287	0.294	0.281	0.302
8	" " 5.	0.181	0.175	0.185	0.185	0.188	0.192	0.174	0.212
9	" " 6.	0.107	0.106	0.107	0.101	0.110	(0.133)	0.099	(0.163)
10	" " 7.	0.065	0.063	0.076	0.061	0.086	(0.112)	0.066	(0.127)
11	" " 8.	0.036	0.033	0.037	0.029	0.047	(0.064)	0.050	(0.085)
12	" " 9.	0.018	0.017	0.022	0.014	0.029	(0.067)	0.034	(0.085)
13	Condenser.	0.013	0.012	0.016	0.014	0.021	0.049	0.039	0.066
14		177	174	171	165	164	165	156	159

The first line in the table gives the number of separate observations upon the average of which each of the following figures in the same column is based. The second line shows the number of first-stage nozzles open during the run; the third, the steam-chest pressure in pounds per square inch by gage; and the fourth, the pressure in the first-stage shell in pounds per square inch absolute. Lines 5 to 13 give the absolute pressures in the shells of the 2d to 9th stages and in the condenser, expressed as fractions of the pressures existing at the same time in the first-stage shell. In the lower right-hand part of the table the pressures were low, as may be seen by noting the first-stage pressures in the 4th line, and the readings were therefore subject to large errors both instrumental and observational.* Pressures of less than 2 pounds absolute are represented by the numbers in parentheses and

*Many of the observations recorded on the starboard engine represent impossibilities, although in general and within their limits of error the observations on the starboard engine lead to the same conclusions as those on the port engine.

have little or no significance; this remark does not apply to the condenser pressures in line 13.

If we follow the horizontal line for any stage except the 1st and 9th we find values which are constant within the range due to possible errors of the gages or accidental variations from perfect steadiness of the steam supply. The values in column VII are rather higher than the average. This would be accounted for by supposing that the value 15.5 for the absolute pressure in the first stage was read too low, as would also seem probable upon comparison with the first-stage and steam-chest pressures in column VI. The figures show, on the whole though not markedly, a tendency to increase toward the right. If the figures in any one line ought, with perfectly exact observations, to have been constant, a gradual increase toward the right would indicate that the gage on the first shell had a zero error and read a little too low.

But disregarding such hypothetical considerations, the conclusion to be drawn from the figures as they stand is very simple and clear. When the pressure in the first-stage shell was cut down by a progressive reduction from 20 to 3 in the number of nozzles open, the pressure in any other shell, except the last next the condenser, was reduced in the same ratio and remained a constant fraction of the pressure in the first shell. It is evident that this relation can not hold exactly at the low-pressure end of the turbine, especially for turbines with many stages and small pressure drops; for though it would usually happen, as it did here, that the vacuum was higher at low than at high powers, the absolute pressure in the exhaust would never, under normal working conditions, vary as 8 to 1, which was the ratio of variation with load in the higher stages in the present case. When the pressure approaches that in the condenser, it must be influenced by it, and the last stage or stages will be partly drowned when the load is very small. We find accordingly that the low pressures toward the right lower corner of the table do not fall as low as would correspond to the fall in the higher stages, so that the reduced values or ratios as given in the table show a tendency to increase toward

the right. But with this exception and for pressures as great as 2 or 3 times the condenser pressure, the rule is satisfied.

RELATION OF THE PRESSURES TO THE STEAM FLOW.

4. It is convenient to have the rule put in terms of the total amount of steam passing through the turbine. The pressure drop through the first-stage nozzles is usually greater than the critical, even at full power, and this is true of the *North Dakota's* turbines. Hence the steam flow through each nozzle depends only on the pressure and steam quality in the steam chest and is independent of the back pressure in the first-stage shell. For a given initial steam state, the total steam flow is therefore proportional to the number of nozzles open. This was confirmed in the above trials, in the runs when the steam was measured. Furthermore, when the steam is dry or superheated, the weight which flows through each nozzle is nearly proportional to the steam-chest pressure, if the temperature in the steam chest is constant and if the pressure drop is greater than the critical. These conditions were approximately fulfilled, and we conclude that the whole amount of steam passing through the turbine was proportional to the product of the number of nozzles open by the absolute steam-chest pressure.

Returning now to our table, we find that if we take the ratio of the absolute pressure in the first-stage shell to the above product which represents the total steam flow, we get—after multiplying by 10^5—the nearly constant series of numbers in line 14 of the table. The numbers have a distinct run, but if we add 2 pounds to all the first-stage pressures, which is a not impossible zero error for the gage in question, the run vanishes and the variations become irregular. The first-stage pressures and all the other pressures except in the last one or two shells were therefore nearly proportional to the steam flow and we have, so far as can be concluded from these observations alone, the following approximate RULE: *When the steam flow is cut down by nozzle regulation, the absolute pressure at every point in the turbine except close to the condenser*

is reduced in the same proportion as the steam flow, if the feed to the steam chest is kept of uniform quality and pressure.

If the regulation had been by the main throttle with all the nozzles left open, the pressures for a given steam flow would have been almost the same as with nozzle regulation, but a trifle higher all through. For the wire-drawing would have raised the initial superheat, the volumes would have been larger and the velocities, heat drops, and pressure drops would have had to be a little larger. But the effect on the pressure curve would have been small, though the distribution of the load among the various stages would have been altered.

The observations quoted above may seem hardly sufficient to warrant the setting up of a general rule, but the conclusion drawn from them is confirmed by other experiments, particularly those of Baer* and Gutermuth.† The rule seems to be quite generally true, and though not absolutely exact is still close enough for practical use. Before having access to the *North Dakota's* trial data, about a year ago, the writer made several attempts to solve the problem *a priori,* for he needed the solution for practical use and had not yet seen the 1910 edition of Stodola nor made a thorough search of the literature. The theoretical attacks failed, but an examination of the observations very soon revealed the rule, which was then found to be well established though apparently not very familiar.

APPLICATIONS.

5. The utility of the rule is evident. To investigate the behavior of a proposed turbine at reduced speed, we first make a guess at its new water rate and compute the total steam flow required to give the power needed to drive the propeller at this rate. From this, the flow at full load, and the full-load pressure curve we proceed to find the pressure limits of the stages at the reduced speed, by means of the rule. From our data on stage efficiency at these new steam and blade speeds we lay

*Zeitschrift d. Ver. Deutscher Ingenieure; 53, pp. 1056 and 1102, 1909.
†Zeitschrift d. Ver. Deutscher Ingenieure; 54, p. 82, 1910.

out the expansion line on the Mollier diagram and work back to find the power and the water rate. If our guess was good enough, the work confirms it; if not, it shows how the first estimate of the water rate should be modified, and we repeat the work till the guess and the check agree. The second computation ought to make us certain that our estimated water rate is as near right as the accuracy of our data on stage efficiency will permit.

It thus becomes possible to make fairly close computations of the water rate at cruising speed, and the designer can then re-examine his general scheme of stages and see where it can be improved, if at all. It is not my intention to go into this subject in detail here, for it would require a great deal of space and be the subject for a separate paper, but the rule has other interesting applications.

6. Consider, for instance, the question of thrust balancing. If the drum begins at the right point on the curve of pressure distribution through the stages, the steam thrust at full power may be made to balance the propeller thrust and so relieve the thrust bearing, which is worth while if it can be done without incurring other disadvantages. If the speed is now reduced to ½ and the power to about ⅛, the steam flow will be about 1/5 of that at full power if the water rate goes up 60 per cent., as it will very likely do, though it ought not to. The pressure on the H.P. end of the drum will be reduced in the same ratio as the steam flow, so that the steam thrust aft will be reduced to a little less than 1/5 of that at full power (the exhaust pressure on the other end of the drum will not fall to 1/5). But at ½ speed and ⅛ power, the propeller thrust is ¼ of that at full power. Hence the steam and propeller thrusts will still be in balance, to within something like 1/20 of the whole full-load thrust. If the pressures all through the turbine had to be kept up by closing intermediate valves in order to get satisfactory efficiency, the balance could not be maintained. This drawback to the use of intermediate valves did not exist in the case of the *Salem*, because her turbines had only wheel stages, so that the propeller thrust was

not balanced at any speed but always taken up on the thrust block.

When the computation is made to find where the drum ought to end in order to give a balance at full speed, it will usually be found, at least in short turbines of large diameter like the Curtis, that the drum is too short from the point of view of steam economy. The drum construction permits the use of so many more blade rows per foot of length than are possible on wheels with intermediate diaphragms, that the drum ought to be carried as far up toward the H.P. end as may be. The place to stop is determined by considerations of tip clearance, and depends somewhat on how hard usage the turbine must stand. The difficulty of reconciling the long drum needed to accommodate many blade rows and give good efficiency with the short drum needed to prevent too great a steam thrust aft, can be solved by splitting the drum in two, reversing the L.P. part, and putting it in a separate casing with the exhaust at the forward end. The total length of drum may then be chosen without regard to thrust, and the division made at such a point as to give any desired thrust. The balance once provided at full power will persist, very approximately, at half speed.

7. It is interesting to return to the consideration of the *Salem's* turbines in the light of our rule about the pressure distribution at reduced power. Let us suppose, first, that the intermediate valves are all kept open and that the regulation is by the first-stage nozzles. When the steam flow and r.p.m. are cut down, the heat drop through the first-stage nozzles is much increased on account of the low pressure in the first-stage shell. This increase of steam speed with the decrease of blade speed makes the efficiency of the first stage, which is poor even at full load, very poor indeed, though on account of the greater heat drop, the whole extraction per pound of steam may be greater than at full load. On the other hand, the heat drop in each of the following sets of nozzles is reduced; the steam speed decreases along with the blade

speed, though not so fast, and the efficiency is kept up except in the last stage, which is partly drowned.

Now suppose that the intermediate valves had all been closed, along with the first-stage nozzles and in the same proportion, so as to keep up the original full-load pressure distribution. The heat drops and steam speeds at all the nozzles would have been kept up to their original values,* and the efficiencies of all the stages would have suffered because the blade speeds would have been much too low for the steam speeds. The last stage, instead of being drowned, would have been working on a greater heat drop than at full load, on account of the higher vacuum, and though its efficiency would have fallen, its extraction per pound might have increased.

Any other uniform setting of the intermediate valves would form a case between the two already mentioned, unless the intermediates were closed *more* than the first-stage valves.

Any non-uniform setting would probably be disadvantageous. It can be shown that if two similar stages are working in series and if the speed of revolution is below that required for the best efficiency—as it always is with a marine turbine at reduced speed—the best results are obtained by dividing the load equally between the two. This follows from the parabolic form of the efficiency curve plotted against the ratio of blade speed to steam speed. If, then, a series of similar stages have been designed with such areas as to distribute the load equally, it is highly probable, though not certain, that any sort of regulation of the nozzle areas which artificially produces an uneven distribution of the load will be disadvantageous, if the r.p.m. are too low for maximum efficiency on the given total heat drop available.

Now it is not possible to say offhand which method of regulation would on the whole give the best results; it is merely evident that in principle there may be something to be said on both sides. But the use of the rule as to pressure distribution would enable us to analyze the problem completely if it were

*In reality they would have been increased a very little on account of the greater reheat in each stage.

worth while, and it is obvious that if such a preliminary study had been correctly made with adequate data, the builders would not have fitted intermediate valves which proved upon trial to be needless. Gutermuth concludes from his experiments on a 650-horsepower impulse turbine with 2-row wheels and 4 stages, all provided with valves, that above half load the regulation by first-stage nozzles alone was at least as good as complete regulation, whereas for smaller loads and at reduced speeds, corresponding to marine conditions, the use of the intermediate valves to keep up the pressures improved the water rate. These conclusions, however, might not be valid for a turbine of very different design, especially for one with more stages in which the importance of the first stage was relatively less.

Gutermuth also found that throttle regulation was not so good as regulation by first-stage nozzles; but this conclusion, natural as it seems, might not be so clear for a different turbine. The throttling greatly reduces the heat drop and steam speed into the first stage. Hence if the first stage even at full power and r.p.m. had too great a steam speed, the efficiency of the first stage might be so much improved by the throttling, in comparison with nozzle regulation, as nearly to offset the fact that the total heat drop available was smaller. Moreover, the drying or superheating due to the throttling has by itself a tendency to improve the efficiency all through. The poorer the original full-power efficiency of the first stage, the more likely it would be that throttling would give nearly as good a result as nozzle regulation. Four-row stages, such as the first stage in the *Salem's* turbines, are notoriously inefficient, and in fact no great difference was observed in the *Salem's* trials whether the regulation was by the throttle or by the first-stage nozzle valves.

8. Having now stated the rule as an empirical fact and mentioned a few of its applications, we may conclude by showing that it is rational, though without attempting a formal deduction.

The equation for adiabatic flow of air through a nozzle is

$$M = A \frac{p}{\sqrt{vp}} \times K,$$

where M is the mass per second, A the nozzle area, p and v the pressure and specific volume at entrance to the nozzle, and K a quantity which depends only on the ratio of p to the back pressure beyond the nozzle when the back pressure is above the critical, or is constant for lower back pressures. Suppose that M is reduced while A remains fixed. If the two pressures are both changed in any ratio their ratio to each other remains the same and K does not change. If, furthermore, the pressures change by the same fraction as the flow M, the above equation will be satisfied provided the product pv at entrance to the nozzle does not change its value with the change in p. Now it is well established that the flow of steam through nozzles follows very nearly the same form of equation as the flow of air, and it is also true that the product pv does not change very much as the steam flows through a turbine so that it can not change much at any one point in consequence of changes in the amount of steam admitted to the turbine, if the steam-chest conditions remain fixed. It follows that if the pressures at all points in a turbine with fixed areas do decrease in the same proportion, the steam flow with given terminal conditions must also decrease in that same proportion in order that the equation to which the process is subject shall be satisfied. It does not, of course, follow that the rule given above is the *only* solution of the problem of finding the pressures from the steam flow; but such solutions are usually unique, and practice appears to show that the solution given by the rule is in fact the only one.

TEST OF A FEED WATER HEATER AT THE NAVAL ENGINEERING EXPERIMENT STATION, ANNAPOLIS, MARYLAND.

The object of the test was to determine the B.T.U.s transmitted per square foot of heating surface per hour, per degree Fahrenheit difference of temperature between the steam and the water, under varying conditions of temperature of steam and of water, and with varying proportions of steam and water.

DESCRIPTION OF APPARATUS.

The feed-water heater under test was as shown on Plate I.

The heater consisted of a composition shell or body containing 117 semi-circular ¾-inch tubes, expanded into a composition tube sheet and a cast-steel bonnet; the bonnet being cast so as to form separate cavities or pockets over the two ends of the tubes, the feed water passing through the tubes, while the steam circulates outside of the tubes.

The total heating surface (calculated) for the tubes and tube sheet was 88.22 square feet, while the area through the tubes was .362 square feet.

METHOD OF TEST.

The connections and locations of instruments are shown on Figures 2 and 3. As the station supply of fresh water would not permit of its use in such large quantities as were required for the test, brackish water from the Severn River had to be used, and corrections were made for the difference in specific heats.

The water was supplied to the heater through pipe "N." Its inlet temperature was taken at "D" with a mercurial thermometer reading to half degrees, and at "B'" with a re-

cording thermometer, " B." After passing through the heater the water passed out at " Q," its temperature being taken by a mercurial thermometer " L," and at "M' " with a recording thermometer " M." In order to obtain the correct temperature at " L" it was necessary to place a spiral mixer in the outlet from the heater, to mix the water, as it was found that with out this mixer the temperature at the top and the bottom of the pipe would vary from 10 to 12 degrees at the lower velocities of feed water.

The water after leaving the heater passed through a Venturi tube (not shown) connected to the indicating and recording instrument " A."

In order to maintain a constant feed temperature part of the heated water from the feed heater was by-passed to the tank " P," and mixed with river water supplied by " R"; the tank being used as a suction tank for the pumps supplying water to the heater.

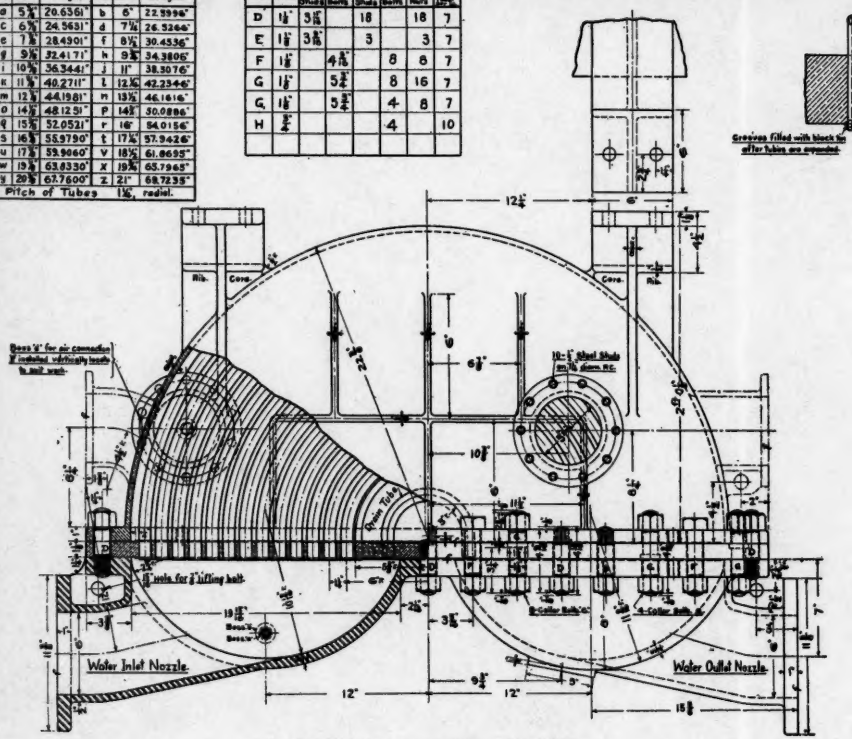
Steam was supplied to the steam side of the heater by the pipe " H" connected to the high-pressure main. As this steam was considerably superheated due to wiredrawing, a low-pressure pipe connection " J" was made to the steam side; and, by the adjustment of the two supplies, a superheat of only a few degrees was held. It was thought better to have some superheat, as then it would be known that dry steam, at least, was being obtained, and a calorimeter would not be necessary. The temperature of the inlet steam was taken at " O," while the pressure was taken by the mercury column " C" at " C'."

The condensed steam was discharged into " S," which was fitted with a gage glass " G." By means of a valve on " K" leading to a condenser (to cool the steam water to prevent reëvaporation), and thence to the weighing tanks, a constant level of water was maintained in " G" to prevent the escape of any steam. The temperature of the condensed steam was taken at " E."

Forty tests in all were conducted, in two groups of twenty each; each group being subdivided into five groups of four each. The two main groups of tests were divided according

Table of Tube Lengths					
Letter	Radius	Length	Letter	Radius	Length.
a	5 5/8"	20.6561"	b	6"	22.5996"
c	6 3/8"	24.5631"	d	7 1/8"	26.5266"
e	7 1/8"	28.4901"	f	8 1/2"	30.4536"
g	9 1/8"	32.4171"	h	9 3/8"	34.3805"
i	10 3/8"	36.3441"	j	11"	38.3076"
k	11 3/8"	40.2711"	l	12 1/8"	42.2346"
m	12 3/8"	44.1981"	n	13 1/2"	46.1616"
o	14 1/8"	48.1251"	p	14 3/8"	50.0886"
q	15 1/8"	52.0521"	r	16"	54.0156"
s	16 3/8"	55.9790"	t	17 1/4"	57.9426"
u	17 3/8"	59.9060"	v	18 1/2"	61.8695"
w	19 1/8"	63.8330"	x	19 3/4"	65.7965"
y	20 3/8"	67.7600"	z	21"	68.7235"
Pitch of Tubes		1 5/16" radial.			

Table of Bolts and nuts.							
Mark	Size	Length		Number			
		Studs	Bolts	Studs	Bolts	Nuts	Nuts
D	1 1/2"	3 5/16"		18		18	7
E	1 1/2"	3 3/16"		3		3	7
F	1 1/2"		4 3/16"		8	8	7
G	1 1/2"		5 3/4"		8	16	7
G	1 1/2"		5 3/4"		4	8	7
H	3/4"				4		10



Note: Heater as drawn for horizontal installation.
Nozzles shown in dash and double dot for vertical installation.

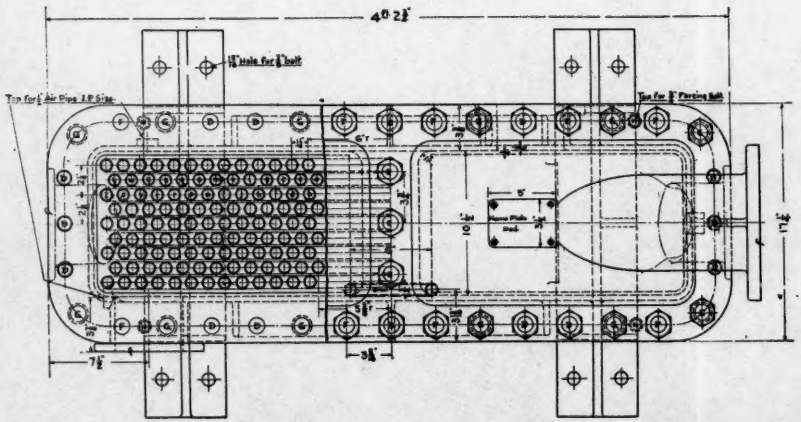
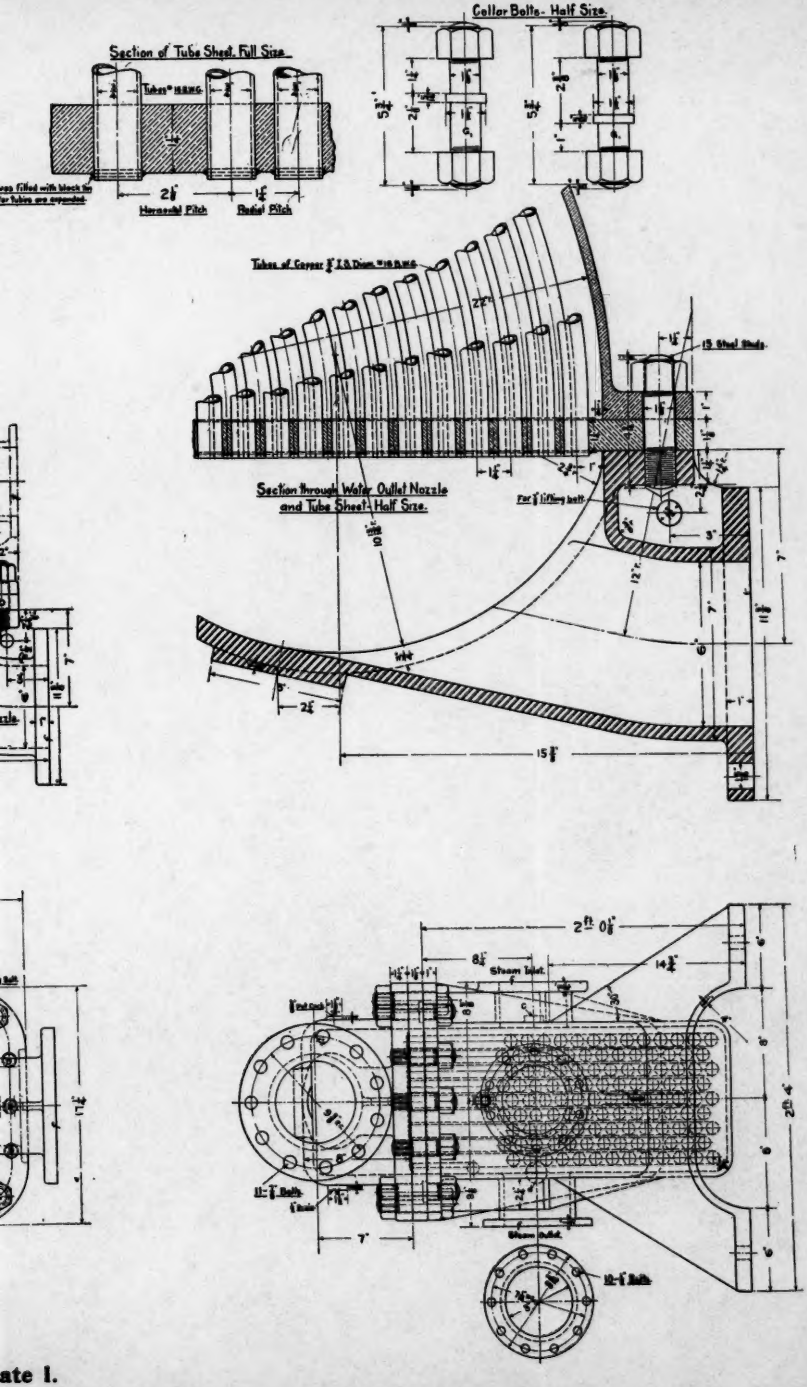


Plate I.


Section of Tube Sheet. Full Size.
Collar Bolts- Half Size.
Tubes of Copper ¾" I.D. Diam. #18 B.W.G.
15 Steel Studs.
Section through Water Outlet Nozzle and Tube Sheet. Half Size.
Steam Inlet.
Steam Outlet.

ate I.

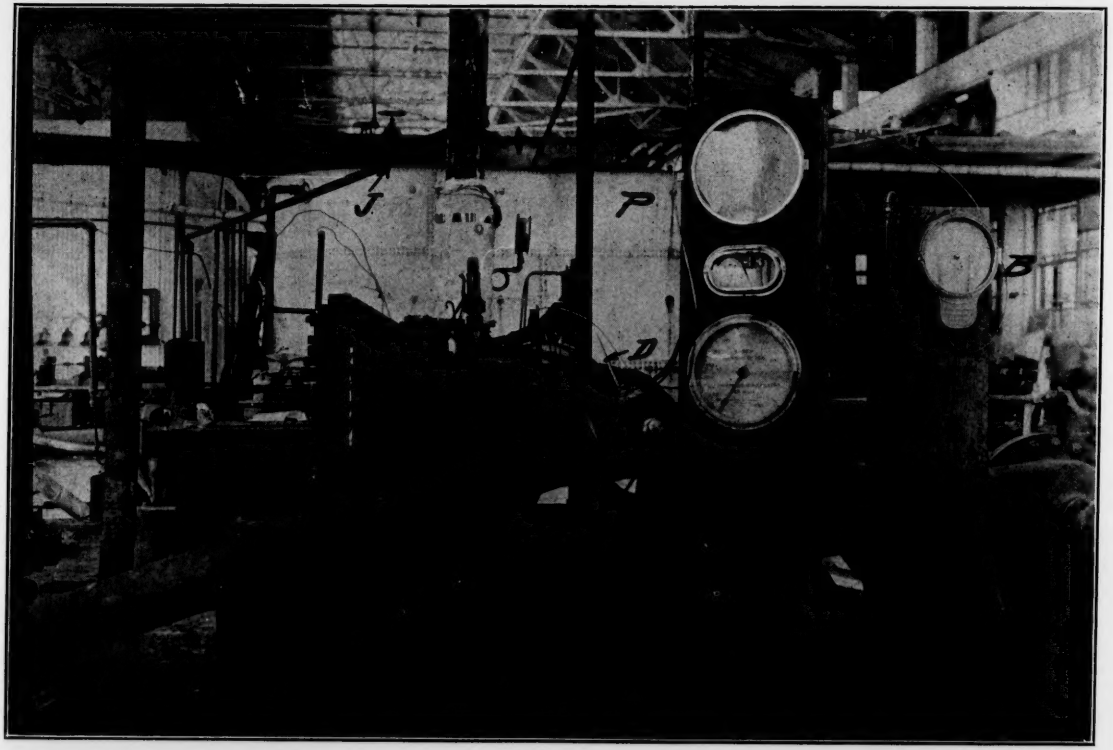


Fig. 2.

Fig. 3.

to the inlet water temperature, that in the first group being about 80 degrees, and that in the second from 130 degrees to 150 degrees F. The subdivision was according to the velocity of feed water and the steam pressure. The velocities taken were 35, 71, 107, 143 and 171 feet per minute, corresponding to a delivery of about 50,000, 100,000, 150,000, 200,000 and 240,000 pounds of water per hour. The steam pressures taken were 5, 10, 15 and 20 pounds gage. The pressure of the feed water in all tests was held from 230 to 250 pounds gage.

All tests were of one-hour duration, and the readings of instruments and gages were taken at ten-minute intervals.

All thermometers were checked during each test with a standard thermometer, and corrections made accordingly.

The scales for weighing water were checked with standard weights before and after the tests.

The Venturi meter was calibrated by pumping the water from the tank "P" (holding 100,000 pounds of water, and which had been carefully calibrated) through the tube, after each change of rate.

RESULTS OF TEST.

The results of tests are given in the accompanying table. Test No. 24 will have to be thrown out, as it is not in agreement with the other test of this group, due to a disarranged water gage. The heater was probably full, or nearly full, of water while the gage showed only the normal water level. This supposition is borne out by the low temperature of the condensed steam shown in column 19 and by the rise of only four degrees in the feed-water temperature.

From the accompanying curves, marked Plate IV, it would appear that the B.T.U. transfers per square foot per hour per degree mean temperature (geometrical) difference, for the two groups, are similar up to about 120 feet per minute velocity of feed water. The higher temperature curve shows a falling off in B.T.U.s transferred per degree difference.

It was noted that the heater as supplied for test was not

furnished with any baffling arrangement to protect the tubes directly under the steam entrance. The erosive effect of the steam on the tubes was very pronounced after two hours' use; and, as the heater had to be opened to replace the rubber gasket on both the steam and water sides, the opportunity was taken advantage of to place a brass scattering plate below the steam opening; and it was found that this plate protected the tubes from any further action of the steam.

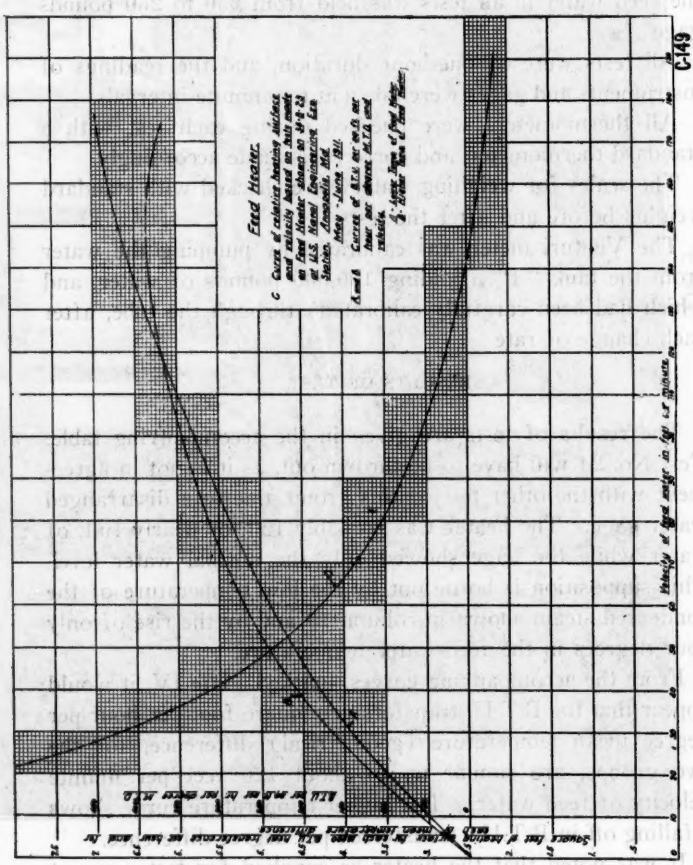


Plate IV.

It was also noted that the bonnet casting was not stiff enough to withstand a pressure of 250 pounds continuously, as the joint between the bonnet and the tube sheet could be seen to open and close with each stroke of the pumps, and on this account could not be kept from leaking.

APPENDIX.

Method of obtaining results:

(Marks and Davis steam tables used for all calculations.)

Column 4, Corrected by Smithsonian Meteorological Tables.

5, From data taken during test.

6, Venturi meter reading corrected for temperature and meter error.

7, Equals column 5 multiplied by .966, the specific heat of river water.

8, From data taken during test.

9, From data taken during test.

10, Column 9 minus column 8.

11, Column 7 multiplied by column 10.

12, From data taken during test.

13, From data taken during test.

14, From data taken during test.

15, Column 14 plus pressure corresponding to barometrical height.

16, From data taken during test.

17, From steam tables, temperature corresponding to pressure in column 15.

18, Column 16 minus column 17.

19, From data taken during test.

20, From formula W (L + (col. 17 — col. 18) + S (col. 18)). Where "W" = col. 13 weight of steam condensed; "L" = latent heat of steam at pressure (col. 15), and "S" = specific heat of superheated steam at pressure, col. 15.

21, Column 20 minus column 11.

Number of test.	Date.	Duration of test (hours).	Barometer, corrected.	Room temperature, degrees F.	Water side.							Steam side.	
					River water pumped through heater, pounds per hour.	Equivalent fresh water pumped through heater, pounds per hour.	Temperature at inlet, degrees F.	Temperature at outlet, degrees F.	Degrees rise.	B.T.U.s absorbed by water, per hour.	Pressure at pump, pounds gage.	Steam condensed, pounds per hour.	Pressure at inlet, inches mercury.
1	2	3	4	5	6	7	8	9	10	11	12	13	14
	May.												
1	6	1	29.66	79.0	49248	47574	84.2	121.8	37.6	1788782	281.7	1917.5	9.08
2	6	1	29.66	79.0	48721	47064	86.3	124.7	38.4	1807258	226.6	1957.5	19.59
3	6	1	29.66	78.0	49690	48001	88.7	128.9	40.2	1929640	236.0	2100.0	29.58
4	6	1	29.59	78.0	49630	47943	90.8	136.7	45.9	2200584	228.9	2416.0	38.34
5	8	1	29.44	71.0	99580	96194	82.8	112.9	30.1	2895439	241.0	3054.5	9.06
6	8	1	29.39	71.0	99550	96165	82.7	113.8	31.1	2990732	238.4	3185.0	19.24
7	8	1	29.30	75.0	99003	95637	83.9	119.3	35.4	3385550	245.3	3624.5	29.08
8	8	1	29.33	73.0	98943	95579	87.1	126.3	39.2	3746697	240.6	4029.0	38.60
9	9	1	29.17	76.0	149625	144538	77.1	97.6	20.5	2963029	236.6	3121.0	9.41
10	9	1	29.14	76.5	149550	144465	77.9	101.7	23.8	3438267	239.3	3635.5	19.06
11	9	1	29.10	76.0	149943	144845	81.1	107.7	26.6	3852877	244.3	4108.0	29.34
12	9	1	29.08	75.0	148372	143810	81.1	114.3	33.2	4774492	258.6	5101.0	38.70
13	10	1	29.24	83.0	200357	193545	79.1	102.2	23.1	4470890	251.9	4605.0	9.00
14	10	1	29.22	83.0	199380	192601	77.2	103.2	26.0	5007626	254.0	5211.0	19.08
15	10	1	29.22	83.0	198583	191831	78.7	108.2	29.5	5659015	257.0	5935.0	29.27
16	11	1	29.31	85.5	199160	192389	79.9	112.3	32.4	6233404	252.6	6590.0	38.30
17	11	1	29.30	85.0	239352	231214	79.7	100.2	20.5	4739887	245.7	4932.0	9.43
21	24	1	29.47	75.5	239420	231280	79.1	103.0	23.9	5527592	234.3	5781.0	19.05
22	24	1	29.46	77.0	239462	231320	80.2	107.4	27.2	6291904	225.7	6602.0	29.24
23	24	1	29.41	73.0	238944	230820	81.7	112.2	30.5	7040010	222.1	7434.0	39.27
18	12	1	29.38	85.0	49385	47706	147.2	171.4	24.2	1154485	249.1	1260.0	8.86
19	12	1	29.37	85.0	49390	47711	144.3	170.2	25.9	1235715	259.1	1362.0	19.09
20	12	1	29.31	85.0	49350	47672	143.7	174.1	30.4	1449229	248.9	1599.0	29.28
	June.												
36	1	1	29.09	82.5	49328	47651	151.8	185.0	33.2	1582013	234.1	1756.0	39.25
	May.												
28	29	1	29.46	86.0	99969	96571	139.2	156.2	17.0	1641707	244.1	1769.0	10.48
29	29	1	29.45	87.0	99495	96112	134.7	154.9	20.2	1941462	241.1	2097.0	20.00
30	29	1	29.44	87.0	99277	95902	133.3	155.6	22.3	2138615	255.0	2329.0	30.71
31	29	1	29.44	87.0	98930	95566	135.6	160.4	24.8	2370037	254.3	2592.0	39.45
32	31	1	29.36	82.0	147376	142365	145.6	161.1	15.5	2206658	253.9	2330.0	9.39
33	31	1	29.36	80.0	146867	141874	144.3	162.5	18.2	2582107	246.0	2754.0	19.48
34	31	1	29.23	81.0	146763	141773	146.4	166.2	19.8	2807105	240.4	3015.0	29.05
35	31	1	29.26	81.0	147631	142612	148.4	170.6	22.2	3165986	247.9	3415.0	39.37
	June.												
37	1	1	29.12	84.0	195842	189183	149.8	163.1	13.3	2516134	249.4	2652.0	9.61
38	1	1	29.12	81.5	197820	191094	147.8	161.6	13.8	2637097	252.4	2814.0	19.50
39	1	1	29.15	81.5	195743	189088	148.1	163.6	15.5	2930864	237.6	3123.0	29.66
40	1	1	29.15	81.5	195604	188953	150.4	168.6	18.2	3438945	242.0	3690.0	38.89
	May.												
24	25	1	29.35	78.5	237447	229374	128.3	132.3	4.0	917496	244.4	990.0	9.89
25	25	1	29.35	78.5	237632	229553	131.0	144.8	13.8	3167831	248.4	3362.0	19.90
26	25	1	29.31	79.8	238008	229916	132.2	148.1	15.9	3655664	250.7	3899.0	30.00
27	25	1	29.30	81.0	237546	229460	129.8	147.0	17.2	3946712	258.7	4213.0	38.86

Steam side.													
Pressure at inlet, pounds absolute.	Temperature at inlet, degrees F.	Temperature at inlet corresponding to pressure, degrees F.	Degrees superheat.	Temperature of outlet steam water to weighing tanks, degrees F.	B.T.U.s rejected by steam.	Radiation from heater, B.T.U.s per hour.	Radiation from heater, calculated from test, B.T.U.s per hour.	Equivalent dry saturated steam condensed, pounds per hour.	Velocity of water through tubes, feet per minute.	Heat transfer per sq. ft. heat'g surf. per deg. differ. (temp. due to pressure, column 17).	Mean temperature difference, geometrical (temperature due to pressure, column 17).	Heat transfer per sq. ft. heat'g surf. per deg. mean temp. diff. (Temp. due to pressure.)	
15	16	17	18	19	20	21	22	23	24	25	26	27	
19.03	227.1	226.0	1.1	218.4	1858878	70096	71088	1934	35.3	148.6	122.3	165.8	
24.20	239.3	238.3	1.0	230.1	1882880	75622	76944	1975	35.0	140.4	131.2	156.1	a
29.09	249.2	248.6	0.6	239.6	2006747	77107	82176	2121	35.7	142.3	137.3	159.3	
33.38	257.6	256.8	0.8	251.7	2285758	85174	86208	2430	35.7	156.1	142.5	175.0	
18.96	232.5	224.9	7.6	216.9	2974313	78874	77520	3091	71.4	237.2	125.9	260.7	
23.89	240.9	237.5	3.4	227.7	3073945	83213	81552	3223	71.4	225.1	139.4	243.2	b
28.67	250.9	247.7	3.2	238.6	3470705	85155	84432	3665	71.0	240.2	143.4	267.6	
33.20	262.8	256.2	6.6	248.3	3836551	89854	91104	4077	71.1	257.2	149.4	284.3	
18.94	231.9	225.0	6.9	214.8	3044573	81544	74832	3165	107.1	233.3	138.1	243.2	
23.66	241.3	237.0	4.3	226.2	3515398	77131	79104	3685	107.1	250.4	143.8	271.0	c
28.70	250.9	247.8	3.1	238.3	3934720	81843	83952	4156	107.4	267.5	152.9	285.6	
33.28	259.5	256.3	3.2	245.2	4864660	90168	88560	5170	106.7	314.7	160.4	337.4	
18.78	249.3	224.6	24.7	211.7	4544932	74042	79824	4723	143.5	354.1	132.8	381.6	
23.72	258.1	237.2	20.9	224.0	5092356	84730	84048	5338	142.8	360.7	149.4	379.9	d
28.72	253.9	247.8	6.1	231.2	5735519	76504	82032	6058	142.3	384.4	154.8	414.4	
33.21	264.5	256.2	8.3	242.8	6316957	83553	85920	6712	142.8	406.1	156.5	451.5	
19.02	235.3	225.3	10.0	212.9	4828428	88541	72144	5020	171.3	375.9	138.1	389.1	
23.83	239.8	237.4	2.4	224.0	5598043	70451	78864	5869	171.3	400.8	144.4	433.9	e
28.83	258.3	248.1	10.2	233.3	6380160	88256	87024	6740	171.6	420.7	156.3	456.3	
33.63	269.6	256.9	12.7	242.8	7143502	103492	94368	7595	171.3	462.1	160.0	498.6	
18.78	225.4	224.6	0.8	218.9	1220038	65553	67392	1268	36.0	178.7	65.1	201.0	
23.80	238.1	236.3	1.8	229.6	1310467	74752	73488	1373	36.0	161.5	78.7	178.0	
28.77	248.9	247.9	1.0	241.1	1525590	76361	78672	1611	36.0	165.9	88.5	185.6	f
33.56	258.4	256.8	1.6	250.3	1664484	82471	84432	1770	36.1	179.7	87.7	204.5	
19.62	229.9	226.9	3.0	218.9	1716355	74648	69072	1786	72.6	221.8	79.0	235.6	
24.29	242.2	238.5	3.7	232.4	2015167	73705	74496	2114	72.2	220.0	93.9	234.4	
29.54	253.3	249.4	2.9	243.7	2219120	80505	79344	2347	72.0	216.6	103.7	233.8	
33.84	258.4	257.3	1.1	251.5	2453688	83651	82272	2609	71.8	228.5	107.3	250.4	
19.04	238.1	225.3	12.8	216.6	2275581	68923	74928	2366	107.2	323.6	72.1	346.9	
23.99	247.9	237.8	10.1	228.3	2665453	83346	80592	2795	106.9	323.1	84.6	346.0	
29.62	250.0	249.6	0.4	238.8	2884137	77032	81120	3050	106.9	316.8	92.1	345.5	
33.71	259.9	257.1	2.8	248.2	3246545	80559	85872	3452	107.6	338.5	96.1	373.4	
19.02	230.5	225.3	5.2	216.9	2579590	63456	70320	2682	142.6	387.3	69.8	408.6	
23.88	240.1	237.5	2.6	225.0	2722680	85583	76128	2855	144.0	344.0	83.4	358.4	
28.89	262.0	248.2	13.8	233.9	3021696	90832	86640	3192	142.5	342.2	93.7	354.6	
33.47	257.5	256.6	0.9	243.5	3521518	82573	84480	3743	142.6	375.8	95.5	408.2	
19.28	226.5	226.0	0.5	180.3	997168	79672	71040	1037	171.6	115.7	102.0	102.0	
24.19	239.3	238.2	1.1	223.9	3254846	87015	77184	3414	172.0	344.1	98.7	363.8	
29.13	250.1	248.6	1.5	235.1	3745126	89462	81744	3958	172.4	364.6	107.1	386.9	
33.47	257.8	256.6	1.2	241.8	4028420	81708	84864	4282	171.9	360.1	123.0	363.7	

Temperature difference over 122 degrees.

Temperature difference below 102 degrees.

22, = (col. 16 — col. 5) 480. The constant "480" was determined by weighing the amount of condensation in the heater due to the room temperature, without any water in the tubes, and reducing this to the B.T.U. radiation loss per hour per degree difference of temperature.

23, By dividing column 20 by the latent heat corresponding to pressure column 15.

24, From $\frac{W}{d \times 60 \times a}$, where "$W$" = weight of water discharged per hour, column 7; "d" = density of water in pounds per cubic foot corresponding to the mean of entrance and discharge temperatures, and "a" the area through tubes, in square feet.

25, $\frac{\text{Column 11}}{(\text{Col. 17} - \text{col. 8}). \times \text{area heating surface (88.22)}}$.

26, From formula:

where t_1 = temperature of outlet water, col. 9; t_0 = temperature of inlet water, col. 8; t_s = temperature of steam, col. 17.

27, $\frac{\text{Column 11}}{\text{Col. 26} \times 88.22}$

It should be noted that in computing columns 25, 26 and 27, the temperature of inlet steam was taken as that due to its pressure, column 17, and not as given in column 16, which shows a slight superheat.

ADDITIONAL TESTS.

Additional tests were run with a constant velocity of water through the heater of 130 feet per minute, a constant steam pressure in the heater of 10 pounds gage, and with variable temperatures of the inlet water. Starting the first test at 80 degrees F. inlet water, and with an inlet temperature of each succeeding test corresponding to the outlet temperature of the preceding test until an outlet temperature of 212 degrees F. was reached, two sets of tests were run; one series

with clear tubes as originally designed, and a similar series with retarders fitted in the tubes.

The apparatus used and the method of conducting the tests were the same as in the foregoing tests with the following exceptions: The heater was thoroughly lagged with 2 inches of magnesia covering, an air cock was attached to the highest point on the heater, and retarders fitted to the tubes for one series of tests.

The air cock was attached when it was found that consistent results could not be obtained for any two consecutive runs. This inconsistency was believed to be due to air collecting in the top of the heater and blanketing a portion of the tube-heating surface, which was borne out by later results after attaching the air cock. This collecting of air is believed to be the cause of the irregularities in the results of the tests previously reported.

Annealed copper strips $\frac{5}{8}$ inch wide and 0.0268 inch thick were twisted into a spiral of 6 inches pitch, and run through all the tubes to act as retarders.

The first test was run with the inlet temperature corresponding to the temperature of the river water (about 80 degrees F.); the outlet temperature from this test being used as the inlet temperature of the second test, and so on until an outlet temperature of 208 degrees was reached. The tests could be carried no further, as the only feed tank available, with a capacity to handle such quantities of water, was an open one, and temperatures above this point caused the generation of too much steam.

Results of Tests.—The results of the tests are given in the accompanying tables. From these results it would appear that the use of retarders increased the heat transfer per hour per square foot of heating surface about 21.6 per cent. at the higher temperature differences, and about 28.9 per cent. at the lower temperature differences.

The drop in pressure through the heater without retarders was 0.55 inch of mercury, and with retarders 0.98 inch of mercury, measured in each case by a differential mercury gage.

TEST OF FEED-

(With velocity of 131 feet per minute,

Test number.	Date, September, 1911.	Duration of test, hours.	Barometer, corrected.	Room temperature, degrees F.	Water side. River water pumped through heater, pounds per hour.	Water side. Equivalent fresh water pumped through heater, pounds per hour.	Water side. Temperature at inlet, degrees F.	Water side. Temperature at outlet, degrees F.	Water side. Degrees rise.	Water side. B.T.U.s absorbed by water, per hour.	Water side. Pressure at pump, pounds gage.	Steam side. Steam condensed, pounds per hour.	Steam side. Pressure at inlet, inches mercury.	Steam side. Pressure at inlet, pounds absolute.	Steam side. Temperature at inlet, degrees F.
1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16
															WITH
1	20	1	29.97	76.0	178543	172473	75.0	108.2	33.2	5726104	55	5943	19.46	24.28	252.6
2	20	1	29.96	79.0	179933	173815	108.9	135.6	26.7	4640861	50	4870	19.48	24.29	252.2
3	21	1	30.13	74.5	176311	170316	136.3	158.8	22.5	3832110	50	3978	20.10	24.67	267.5
4	21	1	30.11	75.0	178260	172199	158.8	176.5	17.7	3047922	50	3208	19.52	24.37	255.2
5	21	1	30.09	75.0	178357	172293	175.6	189.6	14.0	2412102	50	2665	19.70	24.46	244.5
6	21	1	30.07	77.0	175999	170015	189.5	200.8	11.3	1921170	50	2278	19.80	24.50	262.6
7	22	1	30.09	78.5	175731	169756	200.1	208.1	8.0	1358048	50	1802	19.95	24.45	243.4
															WITHOUT
8	26	1	30.03	81.0	178041	171988	78.2	105.6	27.4	4712471	50	4997	19.29	24.23	246.7
9	26	1	30.03	80.0	175948	169966	104.3	128.8	24.5	4164167	55	4523	19.23	24.20	244.4
10	27	1	30.17	77.0	178356	172292	130.4	149.9	19.5	3359694	50	3744	19.47	24.37	243.9
11	27	1	30.16	71.0	176922	170907	149.9	166.9	17.0	2905419	53	3237	19.41	24 34	243.6
12	27	1	30.08	73.0	175459	169493	167.9	182.3	14.4	2440699	50	2753	19.42	24.31	242.0
13	27	1	30.07	74.0	173111	167225	182.7	194.1	11.4	1906365	55	2321	19 30	24.24	242.2
14	28	1	30.11	74.0	171906	166061	192.8	200.5	7.7	1278670	55	1748	19.20	24.21	245.2
15	28	1	30.10	75.0	171695	165857	200.5	206.8	6.3	1044899	50	1552	19.36	24.29	240.5

WATER HEATER.

and steam pressure of 25 pounds absolute.)

Steam side.											
Temperature at inlet, corresponding to pressure, degrees F.	Degrees superheat.	Temperature of outlet steam water to weighing tank, degrees F.	B.T.U.s rejected by steam.	Equivalent dry saturated steam condensed, pounds per hour.	B.T.U.s unaccounted for.	Component velocity of water through tubes (parallel to axis) feet per minute.	Heat transfer per hour per sq. ft. of heating surface, per degree difference (temp. due to press., col. 17).	Mean temperature difference, geometrical (temp. due to press., col. 17).	Heat transfer per hour per sq. ft. heating surface, per degree mean temperature difference (temp. due to press., col. 17).	Pounds of dry steam condensed per sq. ft. of heating surface.	27 reduced to condensation per sq. ft. of tube surface (86 square feet).
17	18	19	20	21	22	23	24	25	26	27	28
RETARDERS.											
238.44	14.16	231.7	5744979	6028	18875	135.7	397.1	146.19	443.99	68.33	69.33
238.47	13.73	232.0	4705297	4937	64436	137.4	406.0	116.34	452.17	55.96	56.78
239.34	28.16	232.4	3870674	4063	38564	135.4	421.6	91.43	475.10	46.06	46.75
238.65	16.55	231.8	3104702	3258	56780	137.7	432.7	70.80	487.98	36.93	37.49
238.86	5.64	232.1	2564663	2691	152561	138.4	432.2	56.00	488.24	30.50	30.94
238.95	23.65	233.6	2208589	2318	287419	136.9	431.7	43.58	499.70	26.27	26.56
239.13	4.27	232.5	1729902	1816	371854	137.4	394.4	34.98	440.07	20.58	20.88
RETARDERS.											
238.33	8.37	224.4	4852641	5091	140170	130.5	333.6	146.29	365.14	57.71	...
238.26	6.14	229.9	4362660	4577	198493	130.0	355.0	121.70	387.86	51.88	...
238.65	5.25	232.1	3602028	3780	242334	131.8	351.8	98.09	388.24	42.85	...
238.58	5.02	232.6	3112181	3265	206762	131.4	371.4	79.92	412.08	37.01	...
238.51	3.49	233.8	2641448	2771	200749	131.0	391.8	63.19	437.82	31.41	...
238.35	3.85	233.9	2226976	2336	320611	129.8	388.3	49.67	435.06	26.48	...
238.28	6.92	233.0	1681279	1764	402609	129.2	318.7	41.49	349.34	19.99	...
238.47	2.03	233.3	1488772	1562	443873	129.4	311.9	34.71	341.23	17.71	...

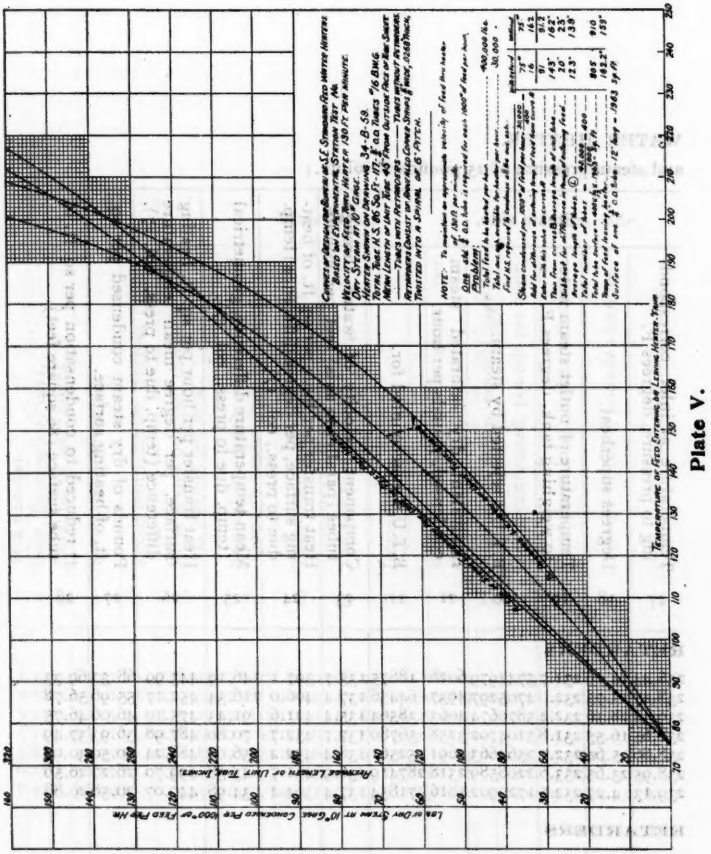


Plate V.

From the curve of heat transfers, B.T.U.s per hour per square foot of heating surface per degree mean temperature difference plotted against the temperature of the inlet water, it would appear that the critical temperature for heat transfer took place at about 190 degrees F., as from this point there is a decided drop in the ratio of transfers.

From the data obtained on these tests the curves of design shown on plate V have been derived. These are based on a constant velocity of feed through the heater of 130 feet per minute.

THE SUCCESSFUL OPERATION OF A SYSTEM OF SCIENTIFIC MANAGEMENT.

By Lieut. Frank W. Sterling, U. S. Navy, Member.

The object of this paper is to present the practical working and routine of a system of modern management as applied to a plant, and particularly to describe the planning and production, personnel, routine, forms and cost keeping. The system at this plant has been much simplified since its installation, resulting in a decrease in the non-productive force, simplification and reduction in the number of forms used, abandonment of the mnemonic classification of stores (in this respect I am firmly convinced that the mnemonic classification of tools should eventually be abandoned), and numerous other improvements. The mnemonic classification of stores is impracticable where a large variety is carried.

In some cases it was found that functions performed by two men could be done by one, and in addition to reducing the non-productive force a move of forms was avoided, thus decreasing the planning time. All this developed as the planning department force became expert and individually were capable of performing more varied functions.

In comparison with the other plants I have visited this one leaves a very strong impression of thoroughness, simplicity, economy and success. Six weeks study of the working of their system confirms my opinion on all the above points.

The system employed was originally installed by Mr. F. W. Taylor and contains all of the essential elements of Mr. Taylor's system. It has been modified as experience and necessity have dictated and has been simplified to such an extent as to excite most favorable comment. All changes have had Mr. Taylor's acquiescence.

DISTRIBUTION OF PERSONNEL.

Productive force, in % of total of men employed	67
Planning department	5
Draftsmen	15
Clerks, etc., main offices	13

ORGANIZATION.

(Chart I to be read concurrently.)

Superintendent.—Since the position of assistant superintendent has been abolished the superintendent's functions have become very comprehensive, combining general management of planning, shop and production, with general superintendence of the planning for personnel. His executive power is in proportion to his functions, being absolute in the planning department, shops, shipping, employment and to a certain extent in the pay department.

Planning Department.—The function of the planning department is to aid production by allotting the work, routing and laying it out, designating the tools to use, speeds, feeds, cuts, etc., to take on the machine, drawing and inspection cards numbers, and any other data that will facilitate production.

Since the system was installed in this plant the planning department has been gradually reduced until no further reduction is possible. The smallness of this department (7% of the size of the productive departments) impresses itself at once. This reduction has been made possible by gradually simplifying the original system, although the advocates of this system consider it a paragon of simplicity.

Comparing the planning department with one other very successful one in the same city, we find that, whereas this one has been reduced to 7% the other remains at about 35% of the productive shop force. The fact that both plants have been very successful induces thought. Even in scientific management there are, apparently, many ways of arriving at the same result, and the old arbitrary methods of the foremost

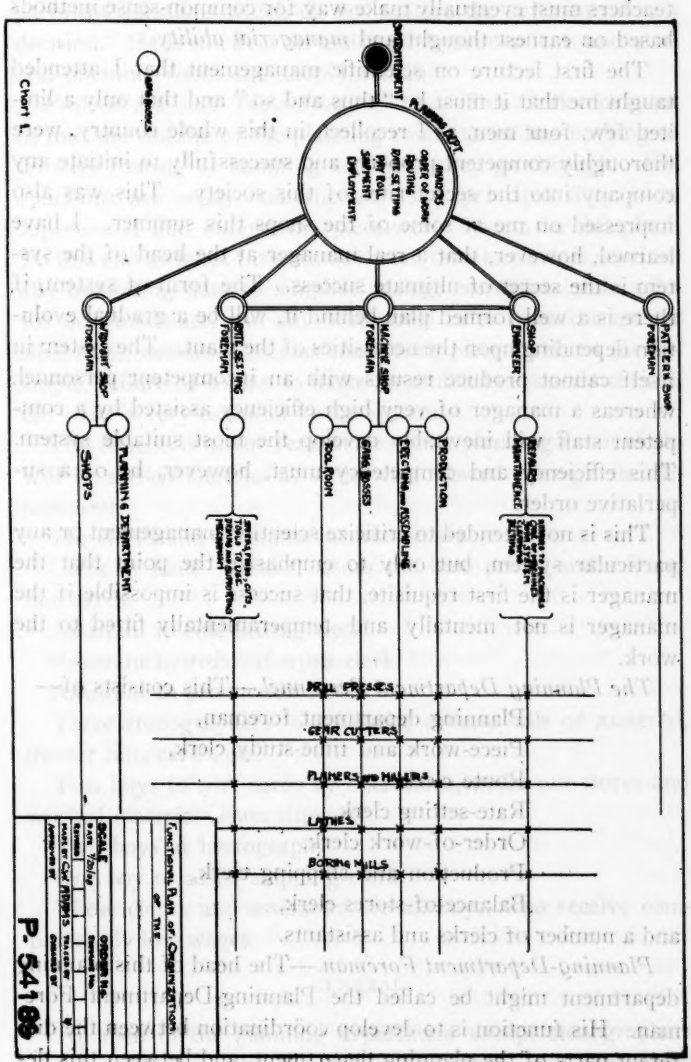


Chart I.
SUPERINTENDENT
PLANNING DEPT.
MINUTES
ORDER OF WORK
ROUTING
RATE SETTING
PAY ROLL
EQUIPMENT
EMPLOYMENT
TIME BOOKS
PATTERN SHOP FOREMAN
SHOP DRAWER
MACHINE SHOP FOREMAN
RATE SETTING FOREMAN
EQUIPMENT SHOP FOREMAN
SHOTS
PLANNING DEPARTMENT
TOOL ROOM
GANG BOSSES
ERECTING and ASSEMBLING
PRODUCTION
REPAIRS and MAINTENANCE
DRILL PRESSES
GEAR CUTTERS
PLANERS and MILLERS
LATHES
BORING MILLS
Functional Plan of Organization
of the
SCALE
ORDER No.
Drawing No.
DRAWN BY
TRACED BY
CHECKED BY
APPROVED BY
P. 54189

Chart I.

teachers must eventually make way for common-sense methods based on earnest thought and *managerial ability*.

The first lecture on scientific management that I attended taught me that it must be "thus and so," and that only a limited few, four men, as I recollect, in this whole country, were thoroughly competent properly and successfully to initiate any company into the secret rites of this society. This was also impressed on me at some of the shops this summer. I have learned, however, that a real manager at the head of the system is the secret of ultimate success. The form of system, if there is a well formed plan behind it, will be a gradual evolution depending upon the necessities of the plant. The system in itself cannot produce results with an incompetent personnel, whereas a manager of very high efficiency assisted by a competent staff will inevitably develop the most suitable system. This efficiency and competency must, however, be of a superlative order.

This is not intended to criticize scientific management or any particular system, but only to emphasize the point that the manager is the first requisite, that success is impossible if the manager is not mentally and temperamentally fitted to the work.

The Planning Department Personnel.—This consists of—

Planning department foreman,
Piece-work and time-study clerk,
Route clerk,
Rate-setting clerk,
Order-of-work clerk,
Production and shipping clerk,
Balance-of-stores clerk,

and a number of clerks and assistants.

Planning-Department Foreman.—The head of this planning department might be called the Planning-Department Foreman. His function is to develop coördination between the different parts of the planning department, and between this department as a whole and the shops, in such manner as to facilitate production. He decides all shop and planning-de-

partment questions which do not require the superintendent's decision. He is in effect an assistant superintendent, and general planning room and shop foreman.

Piece-Work and Time-Study Clerk.—This clerk does not fit into the routine of the planning room, as he has nothing to do with routing or handling orders as they go through. His functions are solely constructive. His principal duty is to superintend and extend the piece-work system in the plant. To this end he takes time studies from time to time, is continually tabulating data to simplify rating, constructs slide rules for new machines, and any other slide rules from the master slide rules, decides questions concerning piece-work, as for example, failure of a man to make an established standard time, etc. He is very essential to the system.

The functions of the remainder of the planning-room clerks will be shown later as an order is followed through the department.

The clerks and assistants mentioned are:

Clerk at issue window, issues the blue prints and instruction cards.

Assistant to rate-setting clerk.

Assistant to order-of-work clerk.

Assistant to production clerk.

Three stenographers to typewrite orders, bills of material, master time cards, etc.

Two boys to sort cards by operations, make out stores and worked material issues slips, etc.

Two boys at hectograph.

One boy to assist shipping clerk.

These clerks and assistants are all boys, who receive comparatively low wages.

GENERAL.

All men in the planning department except stenographers and boys, that is, all having important duties, have come from the shops. They are practical shopmen who have developed into experts at their present positions. The value of their

knowledge of shop methods is obvious, in fact in most instances this knowledge is imperative. Furthermore these men are expert not only at their own billets but in all cases are conversant with the work of the others. The system provides for progressive promotion throughout, which produces a versatility among the planning-room force, tending towards coordination.

THE SHOPS.

(a) At the head of the *machine shop,* designated as D. M., is the machine shop foreman. Under him are the (1) gang bosses, who supervise groups of men, (2) the tool room, (3) the erecting and assembling gangs. Under the gang bosses are the workmen in groups.

(b) The *wrought shop,* designated as D. W., is under the wrought-shop foreman. He has three separate groups under him, being, (1) contract-work gangs, who require practically no supervision, (2) piece-work gangs, such as riveters, who require little supervision, and (3) workmen on day work.

(c) The *pattern shop,* designated as D. P., is under the pattern-shop foreman, who handles the men direct.

The various foremen have supervision over the machine tools in their shops.

(d) The *shop engineer* has two different functions:

(1) The repair and maintenance of buildings. He performs any engineering work required about the plant due to alterations or extensions, and generally is responsible for the condition of the plant as the civil engineer of the establishment.

(2) The repair and care of the machinery of the plant, including belting and shafts. He issues all specifications for new machinery, designs jigs and special fixtures, etc.

There is an assistant to the shop engineer.

The wrought shop has its own planning department. This consists of but three men. Most of the preliminary work previous to actual routing is done in the machine-shop planning department; such preliminary work is copying bills of material,

initial routing of materials from source to shop destination, etc. The wrought shop routing is much simpler than that of the machine shop, because a number of items, which may cover pages of the bill of material, are routed through together, the same being assembled making one division, such as a hopper, casing, etc.

OPERATION OF THE SYSTEM.

It is intended to show the routine from the receipt of an order until the shipment of the finished material, and later the cost-keeping methods in the same manner. The charts should be followed closely with the text. The *raison d'etre* of a comprehensive cost-keeping analysis is its close alliance to efficient and economical production. Without accurate cost keeping it is impossible to get a correct analysis of the business.

Without a correct analysis the weak spots cannot be located with certainty, and we are left groping for our troubles and applying the remedies much as a horse doctor would treat a child. *Cost keeping on jobs serves as a clinic by which we get information on treatment for future jobs.*

CREDIT AND ESTIMATE DEPARTMENTS.

An order from a customer may be preceded by a request for an estimate, in which case it is referred to the estimate department. Copies of estimates furnished to customers are filed in this department.

The function of the credit department is to investigate the credit of customers before making contracts. We have no analogy to this in the U. S. Navy.

CLASSES OF ORDERS.

For convenience of handling in the planning department and shops, job orders are divided into five classes. They are:

(1) Engineering orders, which are orders that require new designing, or engineering work. In this case the order number is preceded by a "W" to idenify it. This class of order

is the closest analogy to those received in a navy yard, so it is treated at length later.

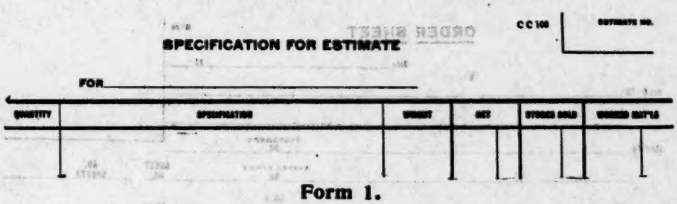
(2) Regular orders, which are orders that do not need new designs as they are for articles already standardized. These can be sent through direct from the order department to the planning department and routed from the original order. These orders include orders for repairs of small parts, and even small orders requiring new designs, where the design can be made by sketch in the order department. The order number in this case is preceded by an "R."

(3) Silent orders, which are orders for silent gears and chain. These are in effect regular orders in their routine handling. The order number is preceded by a "C." This class of work is a specialty of the plant and, a separate account being desired for the class, a separate means of identifying the order is furnished by the prefixed letter.

(4) Worked material orders for store. These orders are for the manufacture of standard articles to place in the stock of the storeroom. They do not emanate from the contract department, as do the previous three classes of orders, but from the balance-of-stores clerk, as a requisition. Approval by the superintendent makes this requisition an order. The order number is preceded by a "Y."

The engineering "W" order, being the most comprehensive in its routine, and being most analogous to navy-yard work, will be treated at length and the variations of routine of the other orders will be pointed out later. The forms used are numbered serially. In many cases the functions of any particular clerk or the data involved in a form will be explained later in the text to avoid undue digression from the main routine of the order.

In most cases of an engineering order "W" an estimate is requested. In these cases Forms 1 and 2 (serial numbers) are used to make up the estimate, and are filed in the estimate department. Data for making up the estimate is obtained from parts of previous similar orders, from stores and worked ma-

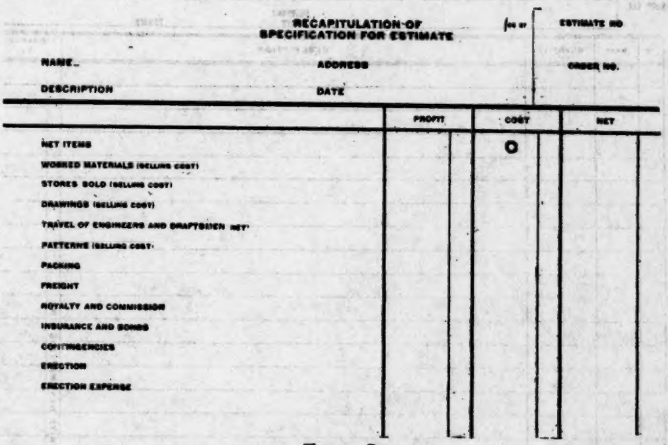

SPECIFICATION FOR ESTIMATE

CC 108

FOR

QUANTITY | SPECIFICATION | WEIGHT | NET | STORES SOLD | STORES RET'D

Form 1.


RECAPITULATION OF SPECIFICATION FOR ESTIMATE

NAME

ADDRESS

DESCRIPTION

DATE

ESTIMATE NO

ORDER NO.

	PROFIT	COST	NET
NET ITEMS		O	
WORKED MATERIALS (SELLING COST)			
STORES SOLD (SELLING COST)			
DRAWINGS (SELLING COST)			
TRAVEL OF ENGINEERS AND DRAFTSMEN NET			
PATTERNS (SELLING COST)			
PACKING			
FREIGHT			
ROYALTY AND COMMISSION			
INSURANCE AND BONDS			
CONTINGENCIES			
ERECTION			
ERECTION EXPENSE			

Form 2.

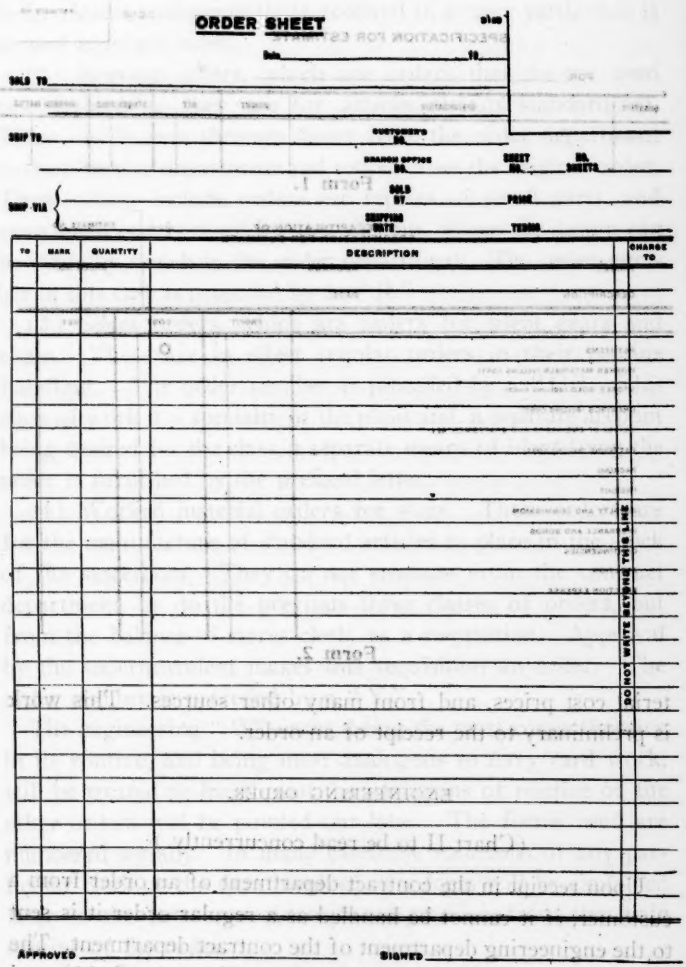
terial cost prices, and from many other sources. This work is preliminary to the receipt of an order.

ENGINEERING ORDER.

(Chart II to be read concurrently.)

Upon receipt in the contract department of an order from a customer, if it cannot be handled as a regular order it is sent to the engineering department of the contract department. The customer's order is here transferred to Form 3 (B126) and sent to the drafting room where necessary drawings are made.

Drafting Room.—This department, designated by symbol D. D. (department drawing) with a personnel of about 15% of the total, prepares all drawings for this plant. The personnel consists of one chief draftsman, with an assistant chief


ORDER SHEET

Date ____ 19__

SOLD TO

SHIP TO

CUSTOMER'S NO.

BRANCH OFFICE NO.

SHEET NO.

NO. SHEETS

SHIP VIA

SOLD BY

PRICE

SHIPPING DATE

TERMS

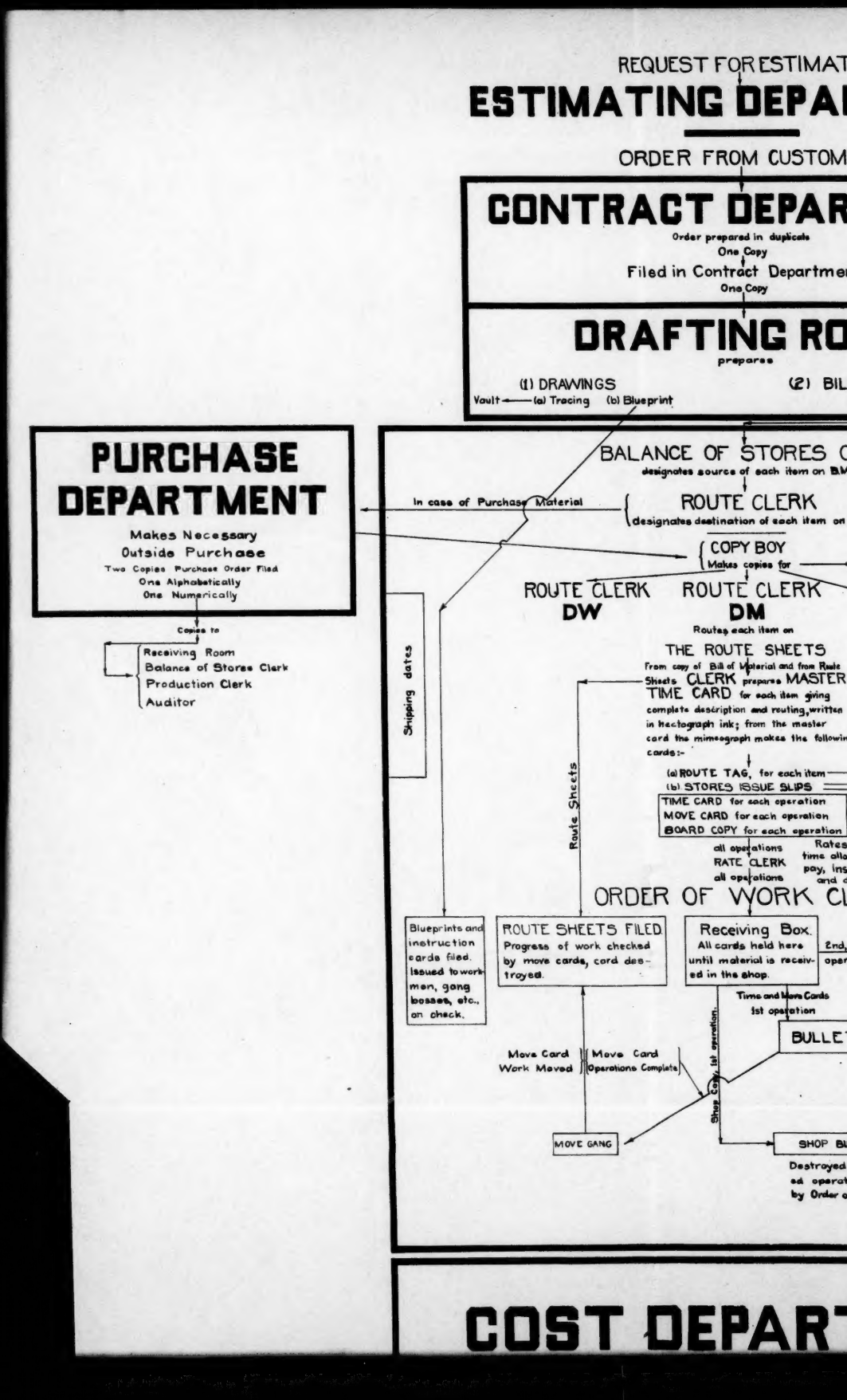
TO	MARK	QUANTITY	DESCRIPTION	CHARGE TO

DO NOT WRITE BELOW THIS LINE

APPROVED ____ SIGNED ____

Form 3.

draftsman, and a number of boss draftsmen. The remaining draftsmen are divided up into groups of three draftsmen, one group under each boss. When an order reaches this department it is turned over to one or more bosses, depending upon


REQUEST FOR ESTIMAT
ESTIMATING DEPA
ORDER FROM CUSTOM
CONTRACT DEPAR
Order prepared in duplicate
One Copy
Filed in Contract Departme
One Copy
DRAFTING RO
prepares
(1) DRAWINGS
Vault — (a) Tracing (b) Blueprint
(2) BIL
PURCHASE DEPARTMENT
Makes Necessary Outside Purchase
Two Copies Purchase Order Filed
One Alphabetically
One Numerically
Copies to
Receiving Room
Balance of Stores Clerk
Production Clerk
Auditor
BALANCE OF STORES
designates source of each item on BM
In case of Purchase Material
ROUTE CLERK
designates destination of each item on
COPY BOY
Makes copies for
ROUTE CLERK DW
ROUTE CLERK DM
Routes each item on
THE ROUTE SHEETS
From copy of Bill of Material and from Route Sheets CLERK prepares MASTER TIME CARD for each item giving complete description and routing, written in hectograph ink; from the master card the mimeograph makes the following cards:-
(a) ROUTE TAG, for each item
(b) STORES ISSUE SLIPS
TIME CARD for each operation
MOVE CARD for each operation
BOARD COPY for each operation
all operations
RATE CLERK
all operations
Rates
time allo
pay, ins
and
Shipping dates
Route Sheets
ORDER OF WORK CL
Blueprints and instruction cards filed. Issued to workmen, gang bosses, etc., on check.
ROUTE SHEETS FILED
Progress of work checked by move cards, card destroyed.
Receiving Box.
All cards held here until material is received in the shop.
2nd
oper
Time and Move Cards
1st operation
BULLE
Move Card
Work Moved
Move Card
Operations Complete
Shop Copy, 1st operation
MOVE GANG
SHOP BU
Destroyed
ed operat
by Order
COST DEPAR

ESTIMATE

EPARTMENT

CUSTOMER

EPARTMENT

duplicate

Department

G ROOM

(2) BILL OF MATERIAL

Original B.M. filed

PLANNING DEPARTMENT

TORES CLERK

ach item on B.M.

LERK

f each item on B.M.

OY

s for

LERK

em on

SHEETS

ial and from Route

ree MASTER

ch item giving

routing, written

the master

akes the following

Original

COST DEPARTMENT
MACHINE SHOP INSPECTOR
WROUGHT SHOP INSPECTOR
ERECTION FLOOR.
FOUNDRY
DRAWING ROOM
PRODUCTION CLERK
SHIPPING ROOM
PLANNING ROOM, etc.

Copies of B.M. destroyed after shipment.

Route Tag remains on item until ready to ship; then it is destroyed.

RECEIVING and STORE ROOMS

MATERIAL from
Purchase
Foundry
Stores
Worked Material Stores

SHIPPING ROOM

or each item

E SLIPS

Stores Issues for D.M.
Stores Issued for C.W.

operation
ch operation
each operation

Clipped together in sets, by operations

ons
ERK
ons

Rates Cards, notes time allowed, rates of pay, instruction card and drawing numbers.

Notification of Arrival of Stores

ORK CLERK

g Box.
d here
is receiv-

2nd, 3rd, etc. operations

Dispatching Box.
Operations not underway - Material in shop.

and More Cards
operation

1st op. Next Cards

BULLETIN BOARD

TIME CLOCK

Time Card

Slips Copy 2nds, 3rds, etc operations

SHOP BULLETIN BOARD

Destroyed when completed operation is checked by Order of Work Clerk.

WORK MAN

Daily wage Statement.

Time of starting and finishing job stamped on card.

Stores Issue D.M. & C.W.

ARTMENT

Copy of B.M. filed.

Balance of Stores Clerk

ORDER FROM CUSTOM

CONTRACT DEPAR

Order prepared in duplicate
One Copy
Filed in Contract Departme
One Copy

DRAFTING RO

prepares

(1) DRAWINGS
Vault ← (a) Tracing (b) Blueprint

(2) BIL

PURCHASE DEPARTMENT

Makes Necessary
Outside Purchase
Two Copies Purchase Order Filed
One Alphabetically
One Numerically

Copies to
- Receiving Room
- Balance of Stores Clerk
- Production Clerk
- Auditor

BALANCE OF STORES
designates source of each item on Bl

In case of Purchase Material

ROUTE CLERK
designates destination of each item on

COPY BOY
Makes copies for

ROUTE CLERK **DW**

ROUTE CLERK **DM**

Routes each item on

THE ROUTE SHEETS

From copy of Bill of Material and from Route Sheets CLERK prepares MASTER TIME CARD for each item giving complete description and routing, written in hectograph ink; from the master card the mimeograph makes the followin cards:-

- (a) ROUTE TAG, for each item
- (b) STORES ISSUE SLIPS
- TIME CARD for each operation
- MOVE CARD for each operation
- BOARD COPY for each operation

Shipping dates

Route Sheets

all operations
RATE CLERK
all operations

Rates time allo pay, ins and d

ORDER OF WORK CL

Blueprints and instruction cards filed. Issued to workmen, gang bosses, etc., on check.

ROUTE SHEETS FILED
Progress of work checked by move cards, card destroyed.

Receiving Box.
All cards held here until material is received in the shop.

End, oper

Time and Move Cards 1st operation

BULLE

Move Card Work Moved

Move Card Operations Complete

Shop Copy 1st operation.

MOVE GANG

SHOP BU

Destroyed ed operati by Order o

COST DEPART

Cha

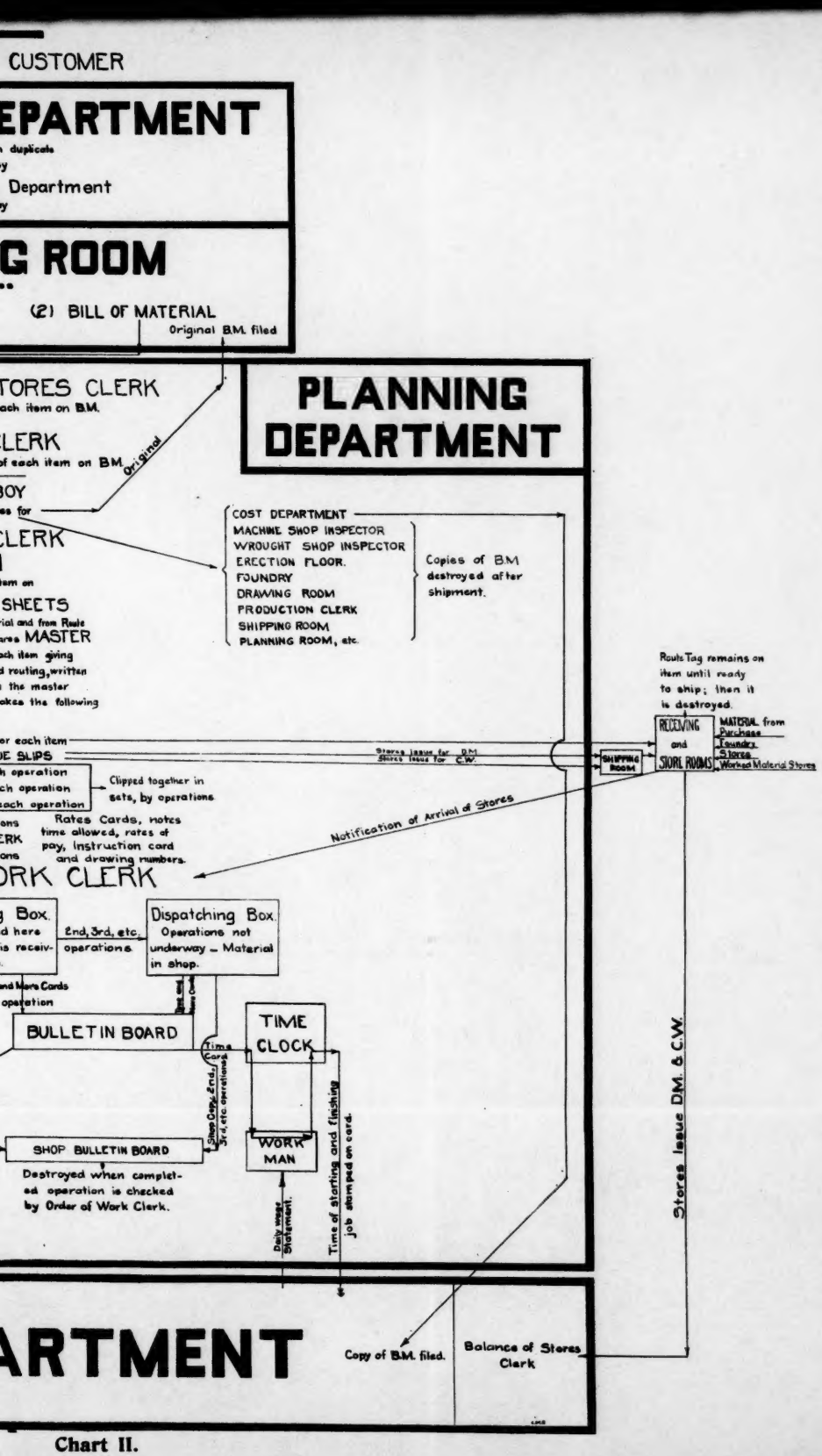
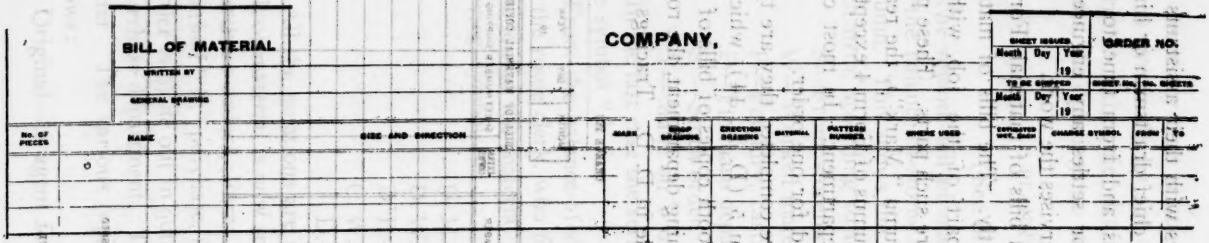

CUSTOMER
EPARTMENT
duplicate
Department
G ROOM
(2) BILL OF MATERIAL
Original B.M. filed
TORES CLERK
ach item on B.M.
LERK
of each item on B.M.
Original
BOY
es for
CLERK
tem on
SHEETS
rial and from Route
ares MASTER
ach item giving
routing, written
the master
akes the following
PLANNING DEPARTMENT
COST DEPARTMENT
MACHINE SHOP INSPECTOR
WROUGHT SHOP INSPECTOR
ERECTION FLOOR.
FOUNDRY
DRAWING ROOM
PRODUCTION CLERK
SHIPPING ROOM
PLANNING ROOM, etc.
Copies of B.M. destroyed after shipment.
Route Tag remains on item until ready to ship; then it is destroyed.
RECEIVING and STORE ROOMS
MATERIAL from Purchase Foundry Stores Worked Material Stores
SHIPPING ROOM
Stores Issue for D.M.
Stores Issue for C.W.
or each item
DE SLIPS
h operation
ch operation
each operation
Clipped together in sets, by operations
Rates Cards, notes time allowed, rates of pay, Instruction card and drawing numbers.
Notification of Arrival of Stores
ORK CLERK
g Box.
d here
s receiv-
2nd, 3rd, etc. operations
Dispatching Box.
Operations not underway — Material in shop.
nd More Cards
operation
BULLETIN BOARD
TIME CLOCK
Time Cards
Shop Copy 2nds, 3rds, etc. operations
SHOP BULLETIN BOARD
Destroyed when completed operation is checked by Order of Work Clerk.
WORK MAN
Daily wage Statement.
Time of starting and finishing job stamped on card.
Stores Issue D.M. & C.W.
ARTMENT
Copy of B.M. filed.
Balance of Stores Clerk

Chart II.



SILL OF MATERIAL

COMPANY,

ORDER NO.

No. OF PIECES	NAME	SIZE AND DIRECTION	MARK	SHOP DRAWING	ERECTION DRAWING	MATERIAL	PATTERN NUMBER	WHERE USED	ESTIMATED WT. EACH	CHARGE SYMBOL	FROM	TO

Forms 4 and 5.

Form 4 is a rough form printed on manila paper. Form 5 is a smooth form printed in hectograph ink on white paper.

REPORT OF FREIGHT RECEIPTS

MO.____ DAY, ____ 191____

SHIPPER	SHIPPING POINT	VIA	DESCRIPTION OF MATERIAL	CHARGE ORDER	FOR PURCHASE ORDER	FREIGHT BILLED				
						CAR NO.	CL.	WEIGHT	RATE	AMOUNT

Form 32.

the size of the order. These bosses with their assistants prepare the necessary drawings. The chief draftsman or his assistant gives necessary instructions and from time to time furnishes necessary information or settles any engineering point involved and, in general, supervises the work.

As the drawings progress rough bills of material, Form 4 (D. D. 15) are written concurrently. The bills of material contain full description of every part of the job, with all necessary dimensions to manufacture such parts. These parts are numbered consecutively (in column "Mark"), the reason for which will appear later. All columns of Form 4 except the last three are filled out in this department. In most cases many sheets of Form 4 are required for one order.

As the rough bills of material are completed they are typewritten in hectograph ink on Form 5 (D. D. 14), which is printed in the same ink and the smooth copies of bills of material, Form 5, are sent to the planning department, the rough copy, Form 4, being retained on file in D. D. Tracings and

COPYING ORDER

CHARGE TO

PHOTOGRAPHIC DEPARTMENT—PLEASE MAKE THE FOLLOWING

THE WRITER SHOULD STATE IF BLUE PRINTS, BLUE LINE PRINTS OR BROWN PRINTS ARE REQUIRED

MONTH	DAY	YEAR 19

PRINTS				BILL OF MATERIAL COPIES		
QUANTITY	DRAWING NUMBER	QUANTITY	DRAWING NUMBER	EXTRA COPIES	SHEET NUMBER	SHIPPING NUMBER

REMARKS

TOTAL NUMBER OF PRINTS	TOTAL NUMBER OF SQUARE FEET	
		SIGNED

Form 6.

On manila paper.

the necessary blue prints are made from the drawings. The prints are made by the photographic department on order, Form 6, of the chief draftsman. The tracings are filed in the vaults, the drawings go to the route clerk to aid routing.

PLANNING DEPARTMENT.

(See diagram, page 182.)

The smooth bill of material is received in this department by the balance-of-stores clerk, who fills out the next to last column. This shows the source of the material to be used for each item, according to the following symbols:

S—From storeroom.
W—Worked material from storeroom.
F—Foundry, castings.
P—Purchase department, on purchase order.
S°—Stock order.
Y—Made on Y order.

In the last case the serial number of the "Y" order follows the symbol "Y."

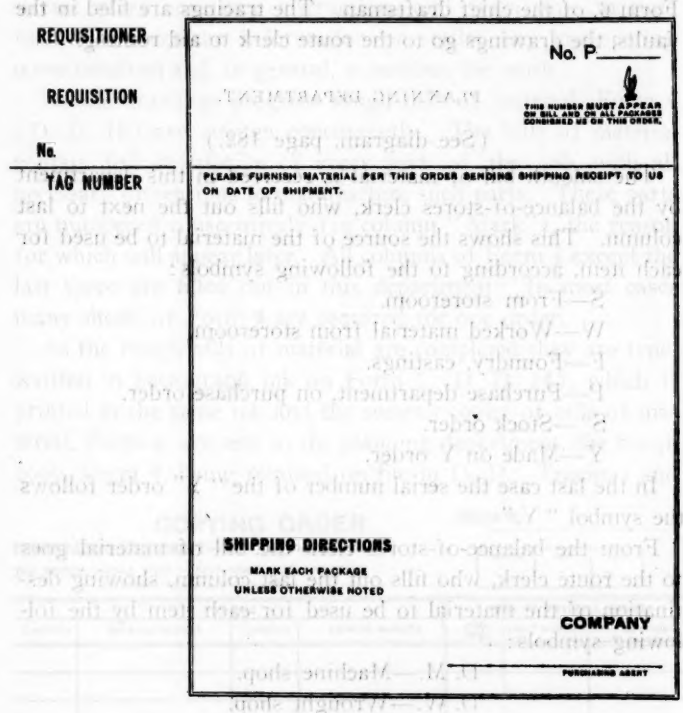
From the balance-of-stores clerk the bill of material goes to the route clerk, who fills out the last column, showing destination of the material to be used for each item by the following symbols:

D. M.—Machine shop.
D. W.—Wrought shop.
D. P.—Pattern shop.
C. W.—Shipping room.
D.—To customer direct.

All the notations are made with copy pencil. The smooth bill of material is now sent to the purchase department, where purchase orders, Form 7 (B1), are made for all articles marked "P," purchase, in next to last column. This purchase order is made out in quintuplicate with a tickler for file in the purchase department. The use of the tickler will be explained later. The various copies of Form 7 are distributed as follows:

Original—to agent from whom purchase is made.

Two copies—filed in purchase department, one numerically and one alphabetically.


REQUISITIONER

REQUISITION

No.

TAG NUMBER

No. P_____

THIS NUMBER MUST APPEAR ON BILL AND ON ALL PACKAGES COMBINED AS ON THIS ORDER.

PLEASE FURNISH MATERIAL PER THIS ORDER SENDING SHIPPING RECEIPT TO US ON DATE OF SHIPMENT.

SHIPPING DIRECTIONS

MARK EACH PACKAGE UNLESS OTHERWISE NOTED

COMPANY

PURCHASING AGENT

Form 7.

This form is in quintuplicate on different colored sheets and Tickler Form 7a.

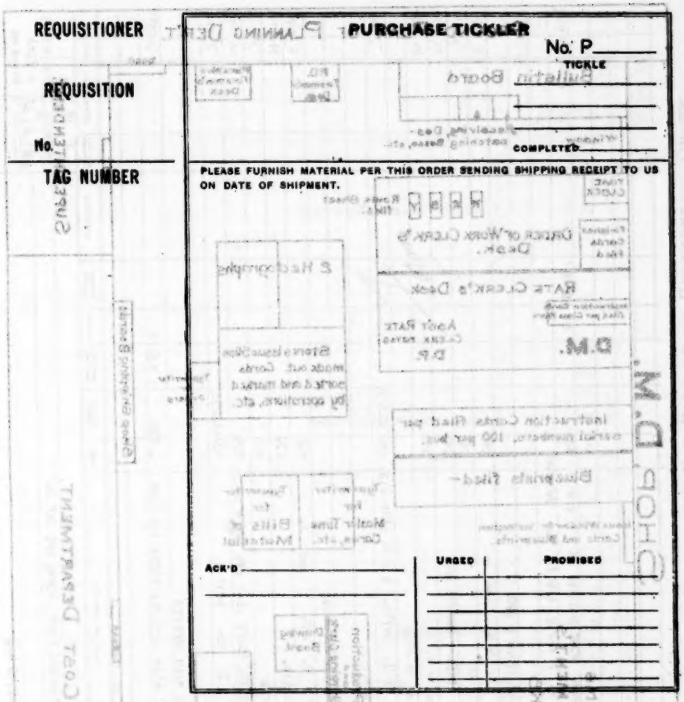
One copy—to receiving room.

One copy—to balance-of-stores clerk.

One copy—to auditor.

Tickler—filed in purchase department tickler.

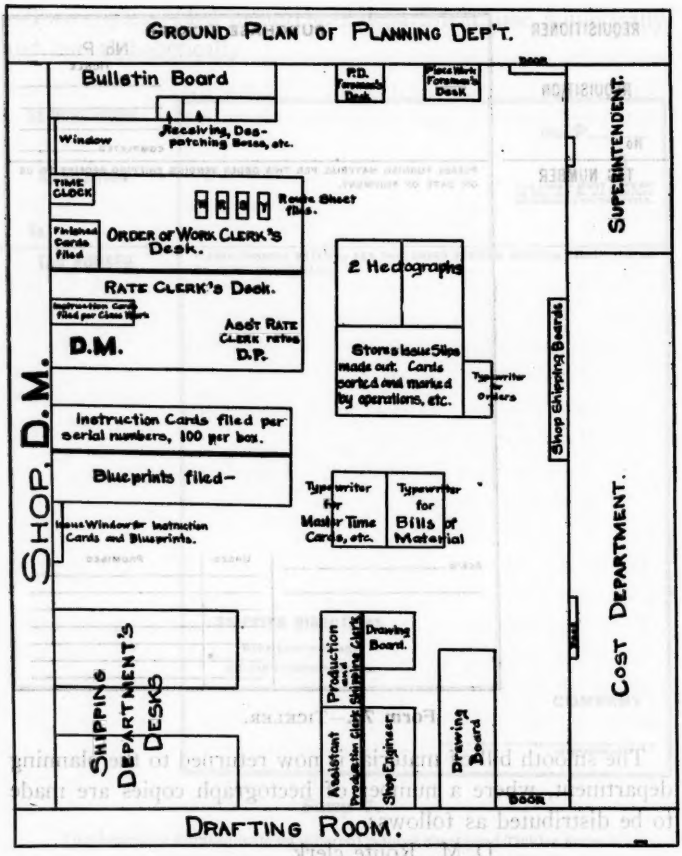
The purchase department enters the purchase order number in copy pencil on the bill of material opposite the letter signifying that the material is to be purchased. This is done so that the various departments can refer to the material by purchase-order number when looking up material due, etc.



Form 7a.—TICKLER.

The smooth bill of material is now returned to the planning department, where a number of hectograph copies are made to be distributed as follows:

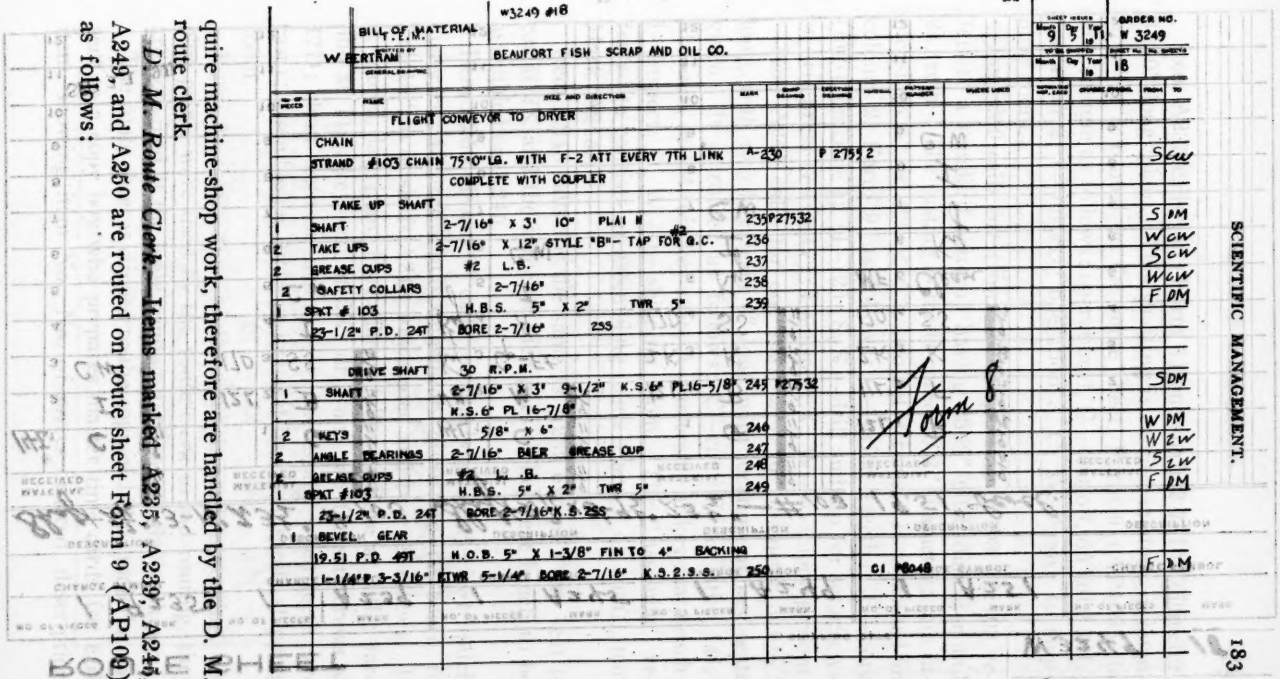
D. M. Route clerk.
D. W. Route clerk.
D. M. Inspector.
D. W. Inspector.
Cost department.
Shipping room, symbol C. W.
Production clerk.
Foundry.
Drawing room.
Main or erection-floor boss.
Planning room.



ARRANGEMENT OF PLANNING DEPARTMENT.

Obviously if there is no foundry or wrought-shop work indicated on the bill of material this copy is omitted. The original bill of material is now returned to file D. D. (drafting room). Form 8 (copy D. D. 14) shows a copy of the 18th sheet of bill of material for order #W3248. This is the copy to the D. M. route clerk.

Items marked in column "to," as D. M. (last column), re-



BILL OF MATERIAL T.E.M. — W3249 #18 — ORDER NO. W 3249

W BERTRAM — BEAUFORT FISH SCRAP AND OIL CO.

NO. PIECES	NAME	SIZE AND DIRECTION	MARK	SHOP DRAWING	ERECTION DRAWING	MATERIAL	PATTERN NUMBER	WHERE USED	FROM	TO
	FLIGHT	CONVEYOR TO DRYER								
	CHAIN									
	STRAND #103 CHAIN	75'0" LG. WITH F-2 ATT EVERY 7TH LINK	A-230		P 2755 2					S CW
		COMPLETE WITH COUPLER								
	TAKE UP SHAFT									
1	SHAFT	2-7/16" X 3' 10" PLAI N #2	235	P27532						S DM
2	TAKE UPS	2-7/16" X 12" STYLE "B"- TAP FOR G.C.	236							W CW
2	GREASE CUPS	#2 L.B.	237							S CW
2	SAFETY COLLARS	2-7/16"	238							W CW
1	SPKT # 103	H.B.S. 5" X 2" TWR 5"	239							F DM
	23-1/2" P.D. 24T	BORE 2-7/16" 2SS								
	DRIVE SHAFT	30 R.P.M.								
1	SHAFT	2-7/16" X 3' 9-1/2" K.S. 6" PL 16-5/8"	245	P27532						S DM
		K.S. 6" PL 16-7/8"								
2	KEYS	5/8" X 6"	246							W DM
2	ANGLE BEARINGS	2-7/16" BAER GREASE CUP	247							W CW
2	GREASE CUPS	#2 L. B.	248							S CW
1	SPKT #103	H.B.S. 5" X 2" TWR 5"	249							F DM
	23-1/2" P.D. 24T	BORE 2-7/16" K.S. 2SS								
1	BEVEL GEAR									
	19.51 P.D 49T	H.O.B. 5" X 1-3/8" FIN TO 4" BACKING								
	1-1/4" P 3-3/16" BTWR 5-1/4"	BORE 2-7/16" K.S. 2 S.S.	250			CI	#8048			F DM

Form 8

Form 8.—This is Form 5 Filled Out For Part of an Engineering Order.

quire machine-shop work, therefore are handled by the D. M. route clerk.

D. M. Route Clerk.—Items marked A235, A239, A244, A249, and A250 are routed on route sheet Form 9 (AP109) as follows:

ROUTE SHEET

SHIPPING ORDER NO. W3249 | SHEET NO. 18

SHIPPING DATE

NO. OF PIECES	MARK	CHARGE SYMBOL	DESCRIPTION	MATERIAL RECEIVED
1	A235		Shaft 2 7/16" x 3'-10"	14L C; I; CW; SEP 11
1	A239		2 3/4" # 103	G; 12L B; 170 SS; I; CW
1	A245		Shaft 2 7/16" x 3'-9"	14L C; 4M M; Ky Cp-Ft; Ky A; I; CW
1	A249		" 2 3/4" — # 103	G; 12L B; 2K K; 170 SS; Ky; I; CW
1	A251		19.51" Level.	12L B; 11L F; 2K K; 170 SS; MF Clean; Ky; I; CW

Form 9.

Item A235 calls for (Form 8) one shaft 2 7/16″ diameter, 3′ 10″ long, plain. This shafting is carried in stock in quantity. It is routed on the route sheet, Form 9 (AP109), in first column as follows: operation 1, C— cut off, on lathe 14L; operation 2, I—inspect; operation 3, C. W.—move to shipping room.

Item 239 calls for (Form 8) a sprocket #103 (#103 refers to a standard pattern) having 23½″ pitch diameter and 24 teeth. The casting is made cored to 2″ diameter; hub dimensions 5″ × 2″ × 5″; two set screws. It is routed in column 2, Form 9, as follows: G—grind (teeth); B—bore (hub), on lathe 12 (L-12); S. S.—drill and tap for set screws on drill press 17 (D-17); I—inspect; C. W—to shipping room.

Item A245 calls for one shaft 2 7/16″ diameter 319½″ long, with two key seats 6″ long with location of key seats given. It is routed as follows: C—cut off, at lathe 14L; M—mill, at 4M, for key seat; A—assemble at key-seat bench; I—inspect; C. W.—to shipping room. The "assemble" referred to consists of assembling items 4 and 5 on shaft, item Mark A245.

Item A249 calls for the same as A239 except a key seat is cut at 2K and the sprocket then goes to key-seat bench to be assembled on A245.

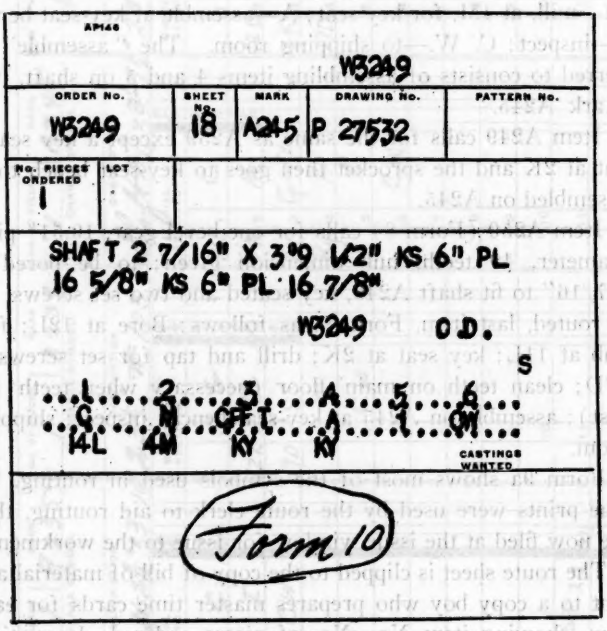
Item A250 (Form 8) calls for one bevel gear, 19.51″ pitch diameter, 49 teeth, hub dimension given; to be bored to 2 7/16″ to fit shaft A249, key seated and two set screws. It is routed, last item, Form 9, as follows: Bore at 12L; face hub at 11L; key seat at 2K; drill and tap for set screws at 17D; clean teeth on main floor (necessary when teeth are cast); assemble on A245 at key-seat bench; inspect; shipping room.

Form 9a shows most of the symbols used in routing. If blue prints were used by the route clerk to aid routing, they are now filed at the issue window for issue to the workmen.

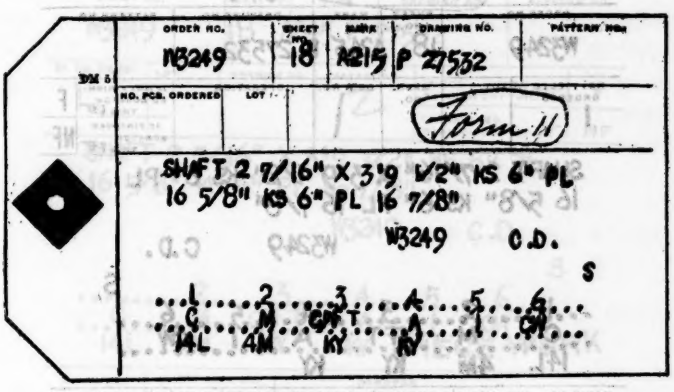
The route sheet is clipped to the copy of bill of material and sent to a copy boy who prepares master time cards for each item, showing item No., No. of pieces ordered, description,

A.	Assemble.	SS.	Setscrew.
B.	Bore.	SAW.	Saw.
BB.	Babbitt.	M.	Mill.
C.	Cut.	CP.	Chip.
CT.	Cut teeth.	FT.	Fit.
F.	Face.	AD.	Assemble and drill.
P–F.	Pin flanges.	Rm.	Ream.
S–F.	Shrink flanges.	P.	Plane.
T–F.	Turn flanges.	Tp.	Tap.
A–F.	Assemble flanges.	OG.	Oil groove.
T.	Turn.	G.	Grind.
RB.	Rough bore.	I.	Inspect.
FB.	Finish bore.	CB.	Counterbore.
R–T.	Rough turn.	FT–JWS.	Fit jaws.
F–T.	Finish turn.	CL.	Clean.
D.	Drill.	Th.	Thread.
L.	Layout.	Cen.	Center.
SP.	Split.	Csk.	Countersunk.
K.	Keyseat.		

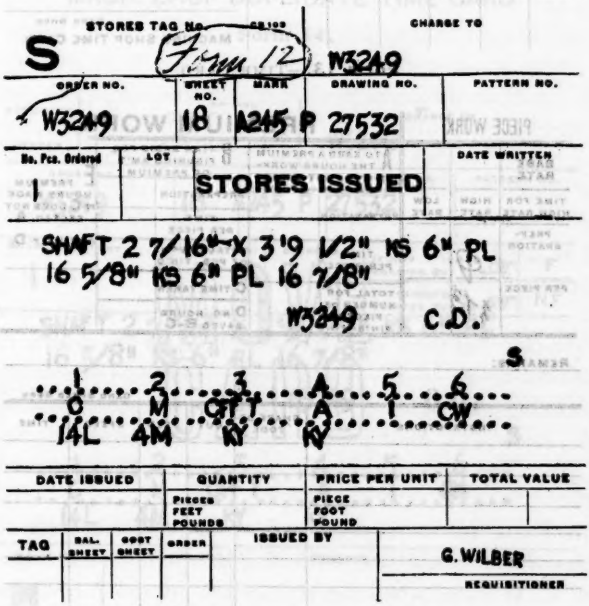
Form No. 9a.—THIS IS NOT A FORM USED IN THE PLANT, BUT SIMPLY A LIST OF THE MNEMONIC SYMBOLS USED IN ROUTING.



MASTER TIME CARD.



Form 11.—Route Tag.



Form 12.—Stores Issue Slip.

RETURNED

ISSUED

Form 13

CHARGE TO W5249

ORDER NO.	SHEET NO.	MARK	DRAWING NO.	PATTERN NO.
W5249	18	A245 P	27532	

NO. PIECES ORDERED	LOT	MACH. NO.	OP'N NO.	OPERATION	IF NOT FINISHED SCRATCH THIS	
1			1		IF FINISHED SCRATCH THIS	F / NF

SHAFT 2 7/16" X 3'9 1/2" KS 6" PL
16 5/8" KS 6" PL 16 7/8"

W5249 C.D.

S

1 2 3 4 5 6
C M CFT A
14L 4W KW KW

MAN'S NO.	NO. PIEC'S FIN. TODAY	TOTAL ALLOWANCE	TIME TAKEN	PREMIUM OR O. T.	MAN'S RATE	MAN'S EARNINGS
DM						

ROUTE SHEET	ROUTE BOARD	PAY SHEET	COST SHEET	MACH. RATE	MACH. CHARGE

Gang Boss

MACHINE SHOP TIME CARD

Form 13.—Time Card.

PIECE WORK

BASE RATE		
TIME FOR HIGH RATE	HIGH RATE	LOW RATE
PREPARATION	9	
PER PIECE	1 13	

PREMIUM WORK

A TO EARN A PREMIUM THE HOURS WORKED MUST NOT EXCEED	
PREPARATION	
TIME PER PIECE	
TOTAL FOR NUMBER OF PIECES FINISHED	

B TIME BASIS FOR FIGURING AM'T. OF PREMIUM	
PREPARATION	
TIME PER PIECE	
TOTAL FOR NO. OF PCS. FIN'D	
C TIME TAKEN	
D NO. HOURS SAVED B-C	

E PREMIUM HOURS MADE IF C DOES NOT EXCEED A ONE-HALF D

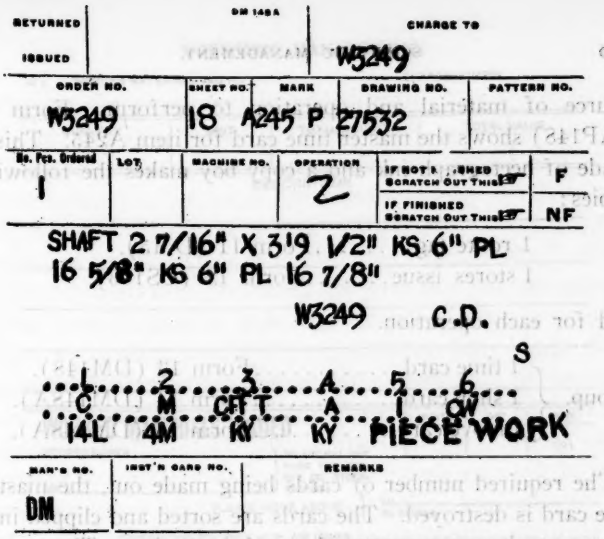
REMARKS:

Gang Speed Boss

INSTRUCTIONS	SHAPE OF TOOL	CUT	FEED	SPEED	TIME

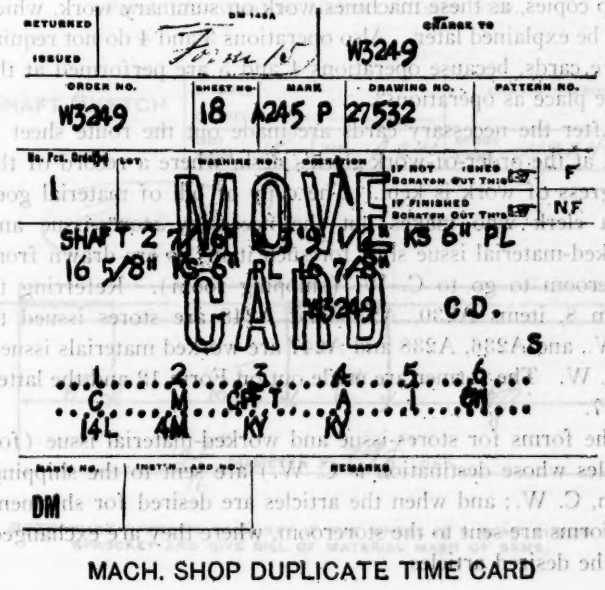
SIGNED

Form 13.—Reverse of Time Card.



MACH. SHOP DUPLICATE TIME CARD

Form 14.



MACH. SHOP DUPLICATE TIME CARD

Form 15.—MOVE CARD.

source of material and operation to perform. Form 10 (AP148) shows the master time card for item A245. This is made of hectograph ink and a copy boy makes the following copies:

1 route tag.....Form 11 (DM5).
1 stores issue.....Form 12 (CS103).

and for each operation.

Group. { 1 time card.....Form 13 (DM148).
1 shop card.....Form 14 (DM148A).
1 move card.....Form 15 (DM148A).

The required number of cards being made out, the master time card is destroyed. The cards are sorted and clipped into groups as above, one group for each operation. The operations are numbered in pencil. The route tag and stores-issue slip are clipped to the cards of the first operation. In the order under consideration operations 1, 2 and 4 do not require shop copies, as these machines work on summary work, which will be explained later. Also operations 3 and 4 do not require move cards, because operations 4 and 5 are performed at the same place as operation 3.

After the necessary cards are made out the route sheet is filed at the order-of-work clerk's desk, where a record of the progress of work is kept. The copy of bill of material goes to a clerk who makes out the necessary stores-issue and worked-material issue slips for such items as are drawn from storeroom to go to C. W. (shipping room). Referring to Form 8, items A230, A237 and A248 are stores issued to C. W., and A236, A238 and A247 are worked materials issued to C. W. The former are made out on Form 12 and the latter on 17.

The forms for stores-issue and worked-material issue (for articles whose destination is C. W.) are sent to the shipping room, C. W.; and when the articles are desired for shipment the forms are sent to the storeroom, where they are exchanged for the desired articles.

Co. WORKED MATERIALS TAG NO. CHARGE SYMBOL

QUANTITY	UNIT	TOTAL WEIGHT	TOTAL VALUE

DESCRIPTION

WORKED MATERIALS ISSUED

STOREKEEPER

MR. _____

DO NOT FILL OUT NAME FOR ORDERS ON STOREKEEPER.

PLEASE ISSUE ABOVE } TO TO BEARER

MONTH	DAY	YEAR
		191

SIGNED BY MAN FOR WHOM W. M. ARE ISSUED

TAG	WORKED MATERIALS SHEET	BALANCE SHEET	

WORKED MATERIALS DESCRIBED ABOVE HAVE BEEN ISSUED

SIGNED BY STOREKEEPER OR HIS REPRESENTATIVE

Form 17.—Worked Material Issue Slip.

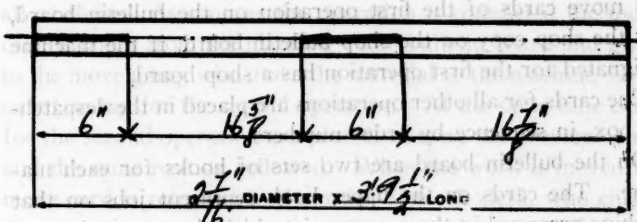
SHAFT SKETCH

NUMBER OF PIECES

CHARGE SYMBOL

SHEET NUMBER 18 | NAME A245 | SHIPPING NUMBER W3249

Form 18



NOTE—MARK OPPOSITE EACH KEY IF THE WHEEL IS A GEAR, PINION, OR SPROCKET AND GIVE BILL OF MATERIAL MARK OF SAME.

The storekeeper, after making the necessary memorandum of issue on his tag (see bottom of Form 12), sends the slips to the cost department, where the issue is entered on the balance-of-stores sheet and cost sheet.

In the case of all shafting items, except plain shaft, a line sketch, Form 18, showing position and character of key seats, accompanies the time card for the milling operation.

All time cards, together with shop copy and move cards, which are clipped to them, go to the rate clerk to be rated. This consists in posting on the back of the time card the necessary data *re* time and pay (explained later), the tools, cuts, feeds, speeds, etc., instruction card and drawing numbers. Some of these are shown on the time cards, front and back, of A245 operation 1. The tags are sent to the receiving room and secured to the material called for by the stores-issue slip, or if the material is not from the store then the tag is secured to the material upon receipt. In the case of A239, A249 and A250, the foundry tag is destroyed when the casting is received and the route tag is substituted. In the case of A235 and A245 slips of stores issued, Form 12, are sent to the storeroom, and tags, Form 11, accompanying them are attached to shafting as soon as it is cut to length. Theoretically the tag is attached before the material leaves the storeroom.

In the meantime the slips for all operations have been placed in the receiving box. As soon as the material is received in the storeroom or receiving room and tagged a notification is sent to the order-of-work clerk, who posts the time and move cards of the first operation on the bulletin board, and the shop copy on the shop bulletin board, if the machine designated for the first operation has a shop board.

The cards for all other operations are placed in the despatching box, in sequence by order numbers.

On the bulletin board are two sets of hooks for each machine. The cards on the upper hook represent jobs on that machine arranged in the sequence in which they are to be performed. The top card represents the job that the man is working on. Cards on the lower hook represent jobs at the ma-

chine, but the sequence in which the jobs are to be performed has not been determined. *The precedence of the slips on the upper hook is regulated by the order-of-work clerk to be in accordance with the order of shipping dates of the various jobs* as obtained from the shipping boards in the planning room.

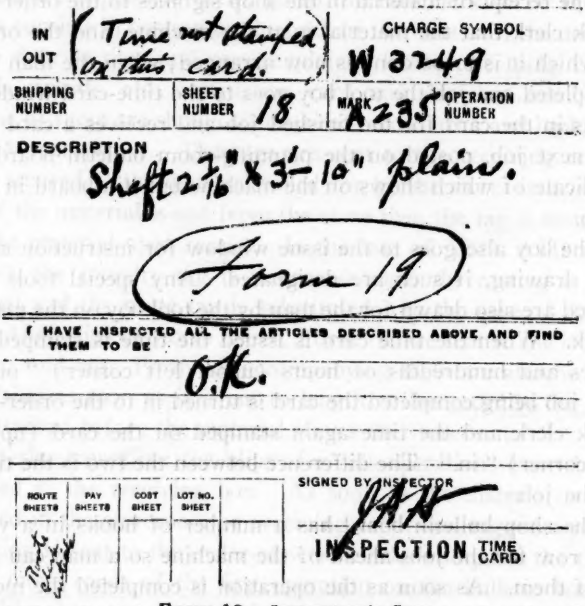
The material is moved from the receiving room to the machine shown on the tag under first operation.

The receipt of material in the shop signifies to the order-of-work clerk that the material is at the machine, and the order in which it is to be done is now arranged; when the man has completed one job the tool boy goes to the time-card window, turns in the card for the finished job and receives a card for the next job, posted on the planning-room bulletin board, a duplicate of which shows on the machine bulletin board in the shop.

The boy also goes to the issue window for instruction card and drawing, if such are designated. Any special tools required are also drawn for the man by the tool boy on the man's check. When the time card is issued the time is stamped in hours and hundredths of hours (upper left corner) "out." The job being completed the card is turned in to the order-of-work clerk and the time again stamped on the card (upper left corner) "in." The difference between the two is the time of the job.

The shop bulletin board has a number of hooks in a vertical row for the jobs ahead of the machine so a man can see all of them. As soon as the operation is completed the move card is taken from the planning-room bulletin board and given to the move gang, who move the material to the machine designated on tag for the second operation. At the same time cards for the second operation are posted, as were the previous ones, and the same routine followed. When the work is moved the move card is returned to the order-of-work clerk, who checks the operation as completed on the route sheet (see Form 9) by drawing a blue vertical line opposite the operation concerned and noting the date. The move card is then destroyed.

When all operations are completed, and after the work is inspected in the shipping room, Form 19 (AP7) is made out and sent to the order-of-work clerk. The blue line on the route sheet is then extended to the bottom of the sheet and the date of inspection is stamped on the route sheet. The order-of-work clerk initials under column "Route Sheet" on Form 19. The route sheet under consideration shows state of work on sheet to be as follows: A235 finished and inspected in C. W.



Form 19.—INSPECTOR'S SLIP.

9/12/11; A239 ready for inspection; A245 went to key-seating bench 9/12/11, being chipped and fitted; A249 went to key-seating bench 9/13/11, and is being assembled to A245; A250 went to main floor 9/13/11, and teeth are being cleaned.

After the time of finishing the job has been stamped upon the time card it is filed by the order-of-work clerk in a box which has compartments for the different classes of machines. These cards are turned into the cost department daily, and the

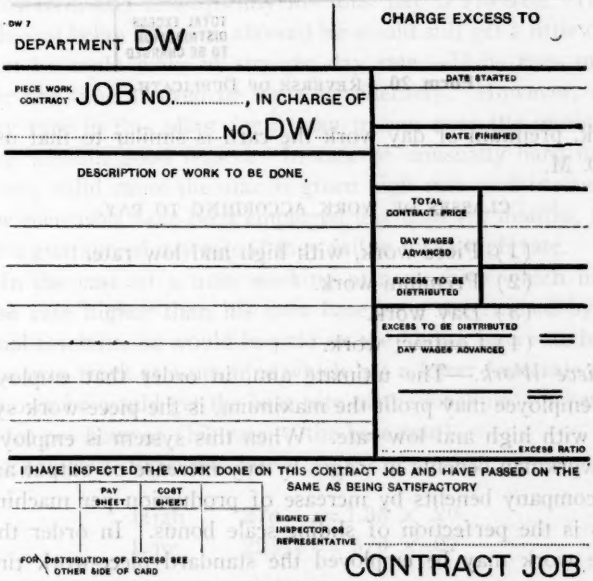
man receives a daily statement of his earnings for the previous day.

Daily all route sheets on which all items have been completed and inspected are taken from the files and sent to the superintendent, and a notification of these jobs and sheet numbers are sent to the production clerk for his information.

When an operation is completed the shop bulletin board copy is checked and destroyed by the order-of-work clerk; all other slips on the board are then moved up a peg.

This is the ordinary procedure for an engineering order as illustrated by Chart II (page 176). In case of work in the wrought shop the routing, etc., is handled by the D. W. route clerk, and a similar procedure takes place. The proposition is much simpler, due to the nature of the work handled.

Considerable contract work is done in this department, and the time card and shop copy are Form 20. In cases of piece


DW 7
DEPARTMENT DW
CHARGE EXCESS TO
PIECE WORK CONTRACT JOB NO. , IN CHARGE OF
NO. DW
DESCRIPTION OF WORK TO BE DONE
DATE STARTED
DATE FINISHED
TOTAL CONTRACT PRICE
DAY WAGES ADVANCED
EXCESS TO BE DISTRIBUTED
EXCESS TO BE DISTRIBUTED
DAY WAGES ADVANCED
EXCESS RATIO
I HAVE INSPECTED THE WORK DONE ON THIS CONTRACT JOB AND HAVE PASSED ON THE SAME AS BEING SATISFACTORY
PAY SHEET
COST SHEET
SIGNED BY INSPECTOR OR REPRESENTATIVE
FOR DISTRIBUTION OF EXCESS SEE OTHER SIDE OF CARD
CONTRACT JOB

Form 20.—Time Card for Contract Work.

In duplicate—original, white; duplicate manila.

DW NO.	WORKMAN'S NAME	DAY WAGES ADVANCED	EXCESS DISTRIBUTION
	TOTAL DAY WAGES ADVANCED		
		TOTAL EXCESS DISTRIBUTION TO BE CHARGED	

Form 20.—REVERSE OF DUPLICATE.

work, premium or day work the card is similar to that used in D. M.

CLASSES OF WORK ACCORDING TO PAY.

(1) Piece work, with high and low rate.
(2) Premium work.
(3) Day work.
(4) Contract work.

Piece Work.—The ultimate aim, in order that employer and employee may profit the maximum, is the piece-work system with high and low rate. When this system is employed the workman benefits in wages by any increase of output and the company benefits by increase of production per machine. This is the perfection of sliding-scale bonus. In order that piece work may be employed the standard piece-work time must be absolutely correct and fair for the average workman. The piece-work rate, once established, cannot be "cut." This

must always be borne in mind, as cutting of rates is invariably followed by discontent among the workmen and spells disaster to the success of the system. These standard times are being added to daily; at present there are more than 10,000 such times established here, and of course many more can be deduced from parts or combinations of parts of these. In cases where these times have been established the man is paid a certain fixed rate per finished piece, the high rate if finished under the minimum time allowed, the low rate if he exceeds this time. These high and low rates are determined from a base rate as follows:

The base rate = the average pay in the neighborhood for work on the class of machine to be operated.

High rate = 1.35 of the base rate (35% bonus).

Low rate = 5/6 × high rate (12½% bonus); but, if the man makes low rate, with the increased time the man's output is lowered, and consequently his total pay is lowered. If he falls just below the time allowed he would still get a little over what he could make at straight day rate. If he runs much over the time allowed he loses considerably. However, it is very rare in this plant for a man to run over the minimum time without good reason. In case of unusually hard metal or any valid cause the man is given high rate, and in case of new men who have been employed less than six months, they get a guaranteed day rate if they fail to make high rate.

In the case of a man working at a machine which has a base rate higher than his own base rate, determined by his usual machine, he would be paid on the higher base rate basis. If a man work at a machine which has a lower base rate than his own he would get the base rate of the machine + a retainer fee per hour of (his rate — the base rate).

Example.—Base rate 20¢

High " 27¢ = 1.35 × .20
Low " 22½¢ = 5/6 × .27

If an 18¢ man work on this job he gets paid on the 20¢ base rate.

If a 22¢ man work on this job he gets paid on the 20¢ base rate, and in addition there is added to his day's earnings 2¢ per hour (called 2¢ retainer) for all his working time. The form for this will be shown later.

Premium Work.—Where a piece-work rate has not been accurately established the work is placed on the premium basis. Such work is chipping and fitting, assembling wheels on shafts, etc. By this system an estimated time allowance is placed on the job. The pay is at a fixed hourly rate. If the man exceed the time allowed he is paid his hourly rate for his *actual working time.* If he does the job in less than the allowed time he is paid for his *actual* working time + ½ (the difference between the time allowed and his actual working time). In other words, he gets a bonus of ½ the time he saves on an estimated time. This gives the fast workman a bonus over the average worker, without doing harm to the slower worker.

Day Work.—Day work is straight pay on a hourly basis. This is applied to movemen, laborers, and for work on which no time can be estimated.

Contract Work.—Work is paid for under this scale as follows: A large job in the truss shop, such as assembling and riveting a tank, is farmed out to a boss workman who will have several men—say a riveting gang—under him. These men all have established day rates. A contract price is placed on the job. This price is distributed in proportion to the hourly rates of the several men. The men are paid daily as follows: Each day the men receive their daily pay according to their hourly rate until the total contract price is paid out. Should the contract be completed before the total contract price is paid out an excess remains to be distributed among the men *pro rata* on their daily rate.

Example.—Suppose a tank to be assembled and riveted, contract price $40.00.

Men employed:

Boss at $5.00 day.
Helper at 3.00 day.
Helper at 2.00 day.

If this job were completed in three days the several amounts paid to date of completion would be:

Boss	$15.00
1st helper	9.00
2d helper	6.00
	30.00
Excess to be distributed	10.00

$$\frac{\text{Excess to be distributed}}{\text{Total days wages paid}} = \frac{10}{30} = \frac{1}{3} = \text{excess ratio.}$$

Excess for each man.

Boss	⅓ × $15	= $5
1st helper	⅓ × $9	= $3
2d helper	⅓ × $6	= $2

See Form 20.

Should the men draw the total contract price before the job is completed the hourly rate is continued as long as the job is in hand. It is a rare case in which the job is not finished before this time. Cases of palpable error in underestimating contract price are subject to readjustment.

Cases of readjustments on pay of men are very rare. Unusually hard metal making a large cutting time, clerical errors, and new men are the only causes that arise. Fairness and contentment mark the working of the system at this plant. During a seven-week period under observation but one case arose where a man got low rate on piece work.

The installation of this system was very gradual. A few of the best workmen were put on piece work, the rate on which had been very carefully established by time study. As soon as it was apparent to the other men that those on piece work were consistently making more than they could make at their day rate, request for piece work came from the men themselves.

Route Clerk.—The route clerk must be a practical shop man, thoroughly conversant with the capacities of machines

and of men and with general shop practices. A tabulated form showing the capacities of the machines is originally prepared, but the route clerk must be conversant with this data to successfully route. He must place the work at the machine which will handle it most efficiently and quickly. He must be in touch with the general condition of work at the different machines so as to distribute the work to best advantage.

Often two operations can be performed simultaneously, such as boring a hub and rough turning the flange of a wheel. He must decide whether to do these as one or two operations, being governed by the work at the various machines, etc. The combinations and possible varieties of routing that pass through his hands are many, and his is a billet that requires knowledge from experience. Routing of items which are often repeated becomes almost automatic.

The route clerk keeps a card index in which he makes a memorandum of the route of any new item. This forms a ready reference for similar items, until such time as he becomes thoroughly familiar with their routing.

Many items, such as gears, etc., are standardized. Tables are prepared for all of these showing the several dimensions. In this case the item is often identified by a number, thus: 23″ sprocket #103. From the standard tables all dimensions can be obtained.

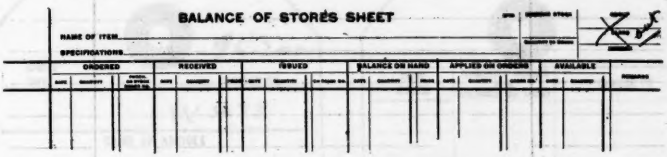
For engineering jobs the blue prints are sent to the route clerk with the B. M. This is necessary to routing, as the interrelation of parts, the dimensions, etc., are thus obtained.

Rate-Setting Clerk, Instruction Cards and Drawings.—The rate-setting clerk determines the class pay for a job. He places the time allowed for preparation, and per piece. Preparation time is allowed on the first piece only where the piece work is used. The base, high and low rate are also placed on the card. An omnimeter having fixed lines on the marker is employed to determine various rates and to determine the pay per piece. The class of pay, allowed times, etc., are found from instruction cards.

Instruction Cards.—There are two sets of these cards, one

filed by class of work for the rate-setting clerk's use, the other filed serially for issue to the workmen. The instruction card numbers and drawing numbers are noted on the time card. When needed by the workmen they are drawn from an issue window on workmen's checks. The cut, feed, speed, machine time, etc., are placed on the time cards. These are obtained from the instruction cards or, if unknown, can be computed by slide rules. A master slide rule is prepared for each class of machine, and slides to fit these master rules are prepared for each machine of the class.

Balance-of-Stores Clerk.—The principal function of the balance-of-stores clerk, as his name implies, is to keep a balance of the stores on hand in the storeroom. This is done on Form 21, balance-of-stores sheet, as follows: Each class of stores has



Form 21.—BALANCE OF STORES SHEET.

its own sheet showing name of item, minimum stock allowed, quantity to order at one time, and columns: ordered, received, balance on hand, applied on orders, and available.

The sheet, Form 21, is kept as follows:

When stores are ordered they are entered with date, quantity and purchase-order number in the first column. Upon receipt a vertical line is entered after the quantity ordered and the same quantity is entered in the second column. Stores are ordered either on requisition, Form 22, on purchase department, if for stores, or by "Y" order, Form 23, if for worked material for store. In either case the requisition or order emanates from the balance-of-stores clerk, when the amount available is less than the minimum stock per his balance sheet. Its approval by the superintendent is authority to purchase in the former case and for manufacture at the plant in the case of a "Y" order.

REQUISITION NO. 57753

DEPARTMENT______ MONTH______ DAY______ 19__

TO THE PURCHASING AGENT;
PLEASE ORDER THE FOLLOWING FOR

TAG NUMBER	DESCRIPTION	NET PRICE PER

WANTED, MONTH______ DAY______ 19__

SIGNED______
REQUISITIONER

STOREKEEPER'S ENDORSEMENT:
NOT IN STORES
OVER
______
STOREKEEPER

APPROVED,

PURCHASE OF______

F. O. B.______ WANTED TO BE SHIPPED, MONTH______ DAY______ 19__ VIA______

SIGNED______
PURCHASING AGENT.

PURCHASE ORDER NO.______ WRITTEN MONTH______ DAY______ 19__ BY______

Form 22.—Requisition for Stores.

Receipts.—In the case of a purchase order, as stated previously, the receiving room is furnished with a copy of the purchase order. When the stores are received in the storeroom, a notification form is sent to the balance-of-stores clerk

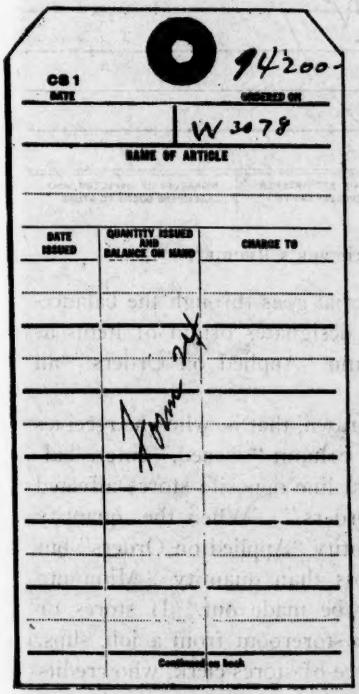
Y-ORDER

SYNOPSIS OF CONTENTS.

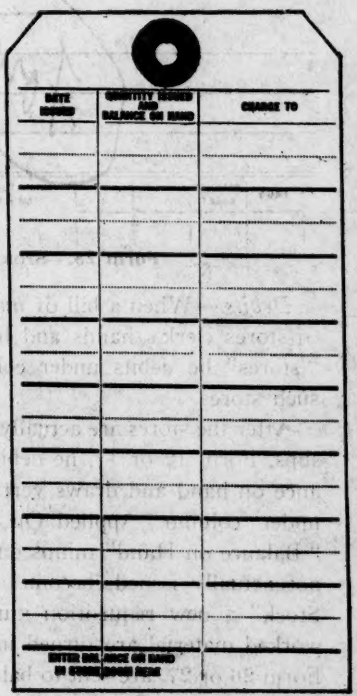
Form 23.

and at the same time the quantity received is entered on the bin tag, Form 24, of the receiving bin.

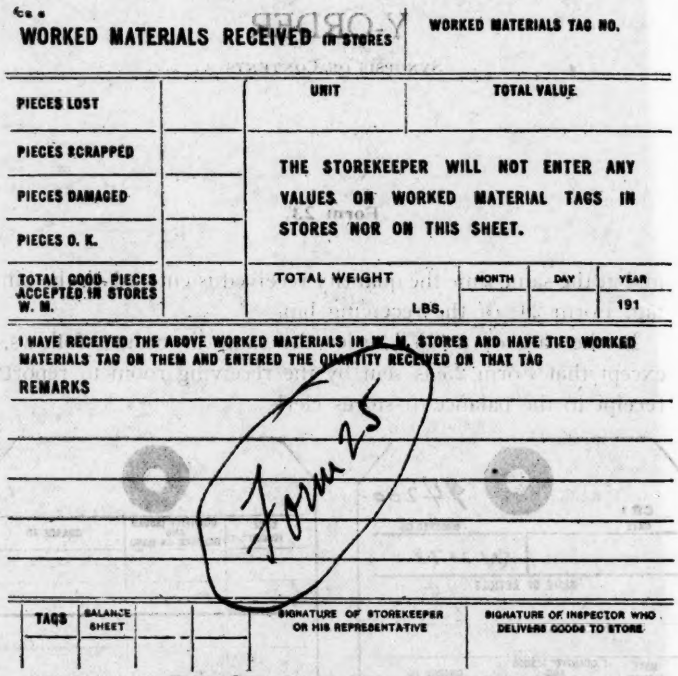
In the case of a "Y" order the same procedure follows, except that Form 25 is sent by the receiving room to report receipt to the balance-of-stores clerk.



Form 24.—BIN TAG.



Form 24.—REVERSE.



WORKED MATERIALS RECEIVED IN STORES

WORKED MATERIALS TAG NO.

		UNIT	TOTAL VALUE
PIECES LOST			
PIECES SCRAPPED		THE STOREKEEPER WILL NOT ENTER ANY VALUES ON WORKED MATERIAL TAGS IN STORES NOR ON THIS SHEET.	
PIECES DAMAGED			
PIECES O. K.			
TOTAL GOOD PIECES ACCEPTED IN STORES W. M.		TOTAL WEIGHT LBS.	MONTH / DAY / YEAR 191

I HAVE RECEIVED THE ABOVE WORKED MATERIALS IN W. M. STORES AND HAVE TIED WORKED MATERIALS TAG ON THEM AND ENTERED THE QUANTITY RECEIVED ON THAT TAG

REMARKS

Form 25

TAGS	BALANCE SHEET			SIGNATURE OF STOREKEEPER OR HIS REPRESENTATIVE	SIGNATURE OF INSPECTOR WHO DELIVERS GOODS TO STORE

Form 25.—STOREKEEPER'S RECEIPT.

Debits.—When a bill of material goes through the balance-of-stores clerk's hands and he designates origin of items as "stores" he debits under column "Applied on Orders," all such stores.

After the stores are actually drawn, that is, when he receives slips, Form 12 or 17, he debits column "Issued," enters balance on hand and draws vertical line opposite stores affected under column "Applied On Orders." When the quantity "Balance on Hand" minus quantity "Applied on Orders" but not actually issued, becomes less than quantity "Minimum Stock" a new requisition must be made out. If stores or worked material are turned into storeroom from a job, slips, Form 26 or 27, are sent to balance-of-stores clerk, who credits under column "Received."

BALANCE OF STORES SHEET

AP23

NAME OF ITEM Bolts

SPECIFICATIONS 7/8" x 3"

MINIMUM STOCK 3000 QUANTITY TO ORDER

ORDERED			RECEIVED			ISSUED			BALANCE ON HAND			APPLIED ON ORDERS			AVAILABLE		REMARKS
DATE	QUANTITY	PURCH. OR STOCK ORDER NO.	DATE	QUANTITY	PRICE	DATE	QUANTITY	ON ORDER NO.	DATE	QUANTITY	PRICE	DATE	QUANTITY	ORDER NO.	DATE	QUANTITY	
8.7	5000	84732														5000	
			8.16	5000						5000	.06	8.8	200	W3094		4800	
												8.8	700			4100	
						8.17	3	R67592		4997						4097	
						8.19	200	W3094		4797							
												8.17	3000			1097	
		Y3246	8.21	50						4847						1147	

Form 21a.—Method of Keeping Balance-of-Stores Sheet.

CR 16 | CREDIT

ORDER NO.	SHEET NO.	MARK	DRAWING NO.	PATTERN NO.

NO. PC'S DEL'D	STORES CREDIT	DATE WRITTEN

NOTE: FINISHED PRODUCT OF DM-DC OR DW MUST NOT BE REPORTED TO STORES UPON THIS FORM

DATE RECEIVED IN STORES	QUANTITY	PRICE PER UNIT	TOTAL VALUE
	PIECES	PIECE	
	FEET	FOOT	
	POUNDS	POUND	

TAG	BAL. SHEET	COST SHEET	ORDER	INSP'TOR	RECEIVED IN STORES BY

Form 26.—STORES CREDIT SLIP.

CR 17 WORKED MATERIALS TAG No. | CREDIT ORDER No.

QUANTITY	UNIT	TOTAL WEIGHT LBS.	TOTAL VALUE

WORKED MATERIALS CREDIT

MONTH	DAY	YEAR
		19

PLEASE CREDIT WORKED MATERIALS ORDER NO. _____

WITH _____

AND CHARGE TO _____

DIRECT LABOR	SHOP EXPENSE	EXPENSE	STORES

I HAVE RECEIVED THE ABOVE WORKED MATERIALS IN WORKED MATERIAL STORES AND HAVE TIED WORKED MATERIALS TAG ON THEM AND ENTERED THE QUANTITY RECEIVED ON THAT TAG

STORES SHEET	BALANCE SHEET	COST SHEET		CARD O.K. SHOP OFFICE	SIG. OF INSPECTOR WHO DELIVERS GOODS TO STORES	SIGNATURE OF STOREKEEPER OR HIS REPRESENTATIVE

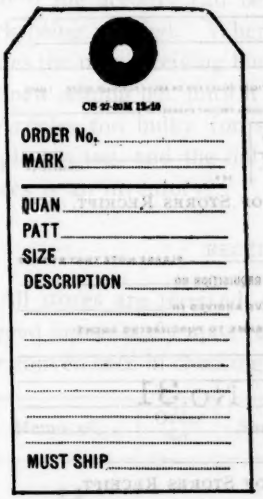
Form 27.—WORKED MATERIALS CREDIT SLIP.

The balance-of-stores sheet, Form 21a, p. 206, under discussion reads minimum stock allowed 3000 8/7/11—5000 ordered on purchase order No. 84732. Same received 8/17/11. 9/18/11—7500 ordered on "Y" order #3246, not yet received—8/21/11—50 turned in on credit slip. 8/17/11—3 issued to regular order 67592 and 8/19/11—200 to engineering order #3094. Balance on hand 8/21/11, 4847. Quantity applied on orders in hand in shops 3700. Quantity available for issue 1147. A new requisition must be made at once, as this quantity is less than minimum. Obviously the 3700 that are reserved for jobs under way in the shops cannot be available for future orders. When performing functions pertaining to stores the balance-of-stores clerk is technically in the cost department.

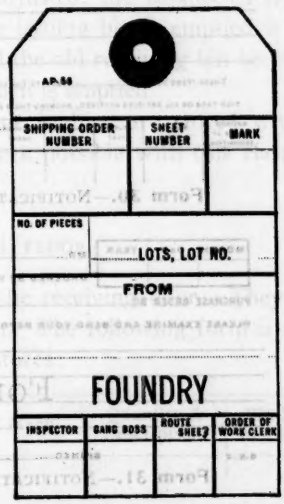
RECEIVING AND STORE ROOM.

Receiving.—Stores may be issued from three sources: (1) Purchased, (2) Worked material made in plant, (3) Foundry, which is a form of purchase.

In case (1) the storekeeper is furnished a copy of the purchase order. Upon receipt of stores he checks up this order and notifies balance-of-stores clerk of receipt of stores by form



Form 28.—Job Order Tag.

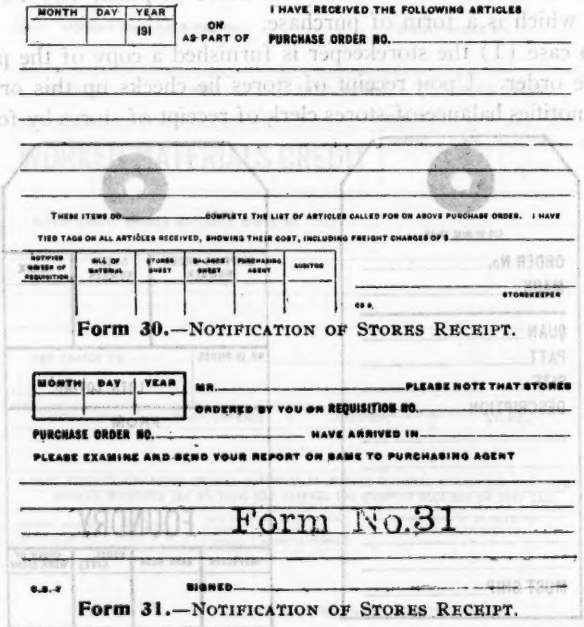


Form 29.—Foundry Tag.

and enters quantity on receiving-bin tag, Form 24, if stores go into bin, or attaches a bin tag and another tag, Form 28, if the stores are destined for a particular job. In the latter case the route tag, when received by the storekeeper, is substituted for the bin tag which is destroyed, and both tags now on the item remain with it to completion in the shops.

In case (2) worked material made in plant, the material is turned into the bin and entered on the bin tag, notification being sent as stated under "Balance-of-Stores Clerk," on Form 25.

In case (3) the procedure follows the second variation of case 1. A bin tag and foundry tag, Form 29, are put on the castings. As in the former case the route tag when received is substituted for the bin tag. Notification of receipts of castings is made to the balance-of-stores clerk, etc., on Form 30, and to the order-of-work clerk for his route-sheet file on Form 31.



Form 30.—NOTIFICATION OF STORES RECEIPT.

Form 31.—NOTIFICATION OF STORES RECEIPT.

Stores drawn for a job order may be returned to the store, in which case the storekeeper sends slip, Form 26 or Form 27, to the balance-of-stores clerk for his stores sheet, and this also serves as a notification to the cost department to credit the job order with stores not used.

ISSUING.

When Form 12 or 17 is presented to the storekeeper, stores are issued, the forms being retained. Tags on *issuing* bins are debited with stores issued. The forms are sent to the balance-of-stores clerk for his records and then go to the cost department for debit on the job order to which they are applied.

STOREKEEPING.

Articles which are carried in quantity are stored in bins. Each item has two bins, a receiving and issuing bin, with a bin tag on each. All receipts go to one bin and all issues come from the other. In this way a check is obtained on the item account. When the issuing bin is empty, if the tag does not read 0 the account can be readjusted, the balance-of-stores clerk being notified. When the issuing bin is emptied it becomes the new receiving bin and the old receiving bin becomes the new issuing bin, until it in turn is emptied.

Articles too bulky for storage in bins are stored with a single bin tag, and the only check possible with this class of stores is an inventory.

RECEIVING FLOOR.

All stores are received on the receiving floor. They are tagged immediately upon receipt. The following form is used for convenience in checking in stores:

Memo. of	Order No.	Sheet & Mach.	Quantity & Description.	Bin No. or Floor.

They are then treated as described under receiving. Freight receipts are reported on Form 32 (see page 177).

INVENTORY.

An inventory is taken annually, but all items are not inventoried simultaneously. An inventory is taken of several classes each month so that at the year's end the storeroom has been entirely covered. This prevents business interruption for this purpose.

PRODUCTION CLERK AND SHIPPING.

The production clerk's function is briefly to get the work out of the shop, coöperating with and notifying the planning-room foreman if there is any probability of delay in shipment.

He is the head of the shipping department and keeps the tickler.

He is informed of the progress of an order at all stages and keeps a record of such progress on Form No. 33 (AP35). He is informed of the receipt of material in the shop by

(1) Manifest in the case of castings.

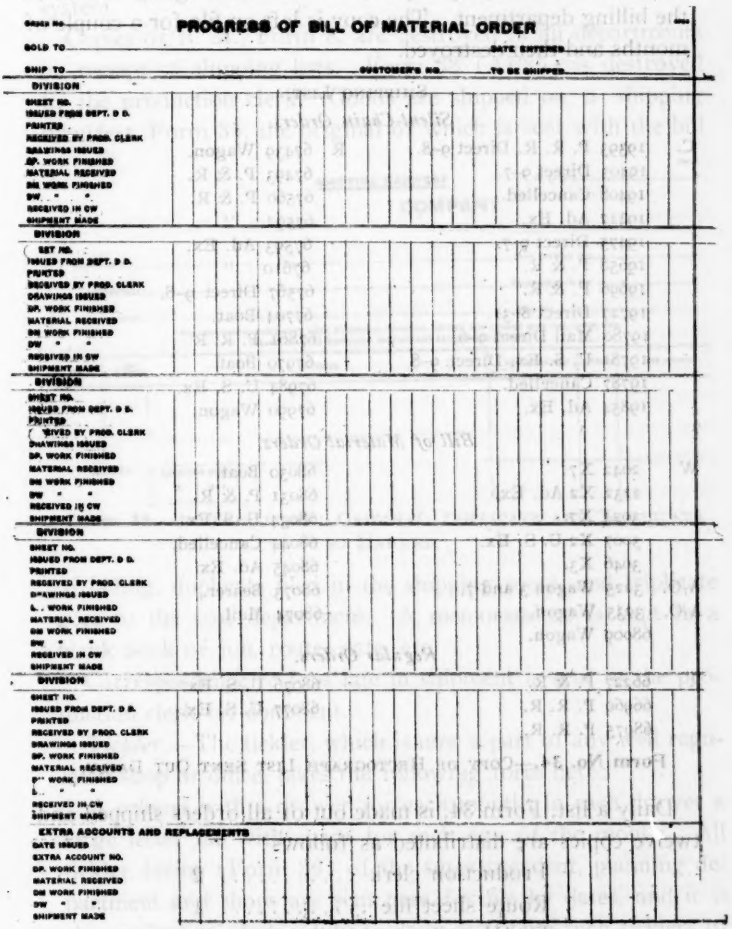
(2) Copy of Form 7 in the case of purchased material.

He keeps a foundry board on which is recorded the dates castings are due, and keeps a record of their receipt on this according to the three-color scheme explained later in the case of D. M. and D. W. shipping boards.

Silent and regular orders come from the contract department to the production clerk to be checked for rush orders so that the material for these jobs can be started before the order is written. In this connection the production clerk notes the date of shipment and enters it on the D. M. and D. W. shipping boards for the planning-department foreman's information. In case of doubt the latter are consulted in fixing this date.

Engineering order shipping dates are not entered on the shop shipping boards. Instead, each department, etc., concerned has a list of same, called a " Bill of Material Shipping

List." This list shows: Firm, order No., date of shipment, preferred and penalty orders, etc. Daily at 9 A. M. the production clerk, planning-room foremen and shop foremen meet in the superintendent's office and discuss the progress of this list.



Form 33.—Progress of Work Kept by Production Clerk.

The original and a copy of each order is sent to the shipping department; the copy goes to the packing room. Goods are packed and shipped as per route previously designated on the order by the production clerk.

When the goods are all checked and packed the copy is returned to the shipping department and the original is sent to the billing department. The copy is left on file for a couple of months and then destroyed.

SHIPPING LIST.

Silent-Chain Orders.

C.	19393	P. R. R. Direct 9–8.	R	67439 Wagon.
	19403	Direct 9–7.		67493 P. & R.
	19408	Cancelled.		67560 P. & R.
	19412	Ad. Ex.		67592 "
	19475	Direct 9–7.		67593 Ad. Ex.
	19658	P. & R.		67610 "
	19696	P. & R.		67567 Direct 9–8.
	19727	Direct 8–31.		67704 Boat.
	19780	Mail Direct 9–6.		67861 P. R. R.
	19781	U. S. Ex. Direct 9–8.		67970 Boat.
	19787	Cancelled.		67984 U. S. Ex.
	19854	Ad. Ex.		67990 Wagon.

Bill of Material Orders.

W	2042	X7.	68030 Boat.
	2232	X2 Ad. Ex.	68031 P. & R.
	3024	X7.	68034 U. S. Ex.
	3007	X2 U. S. Ex.	68044 Cancelled.
	3046	X3.	68045 Ad. Ex.
A/O.	3225	Wagon 3 and 7.	68073 Bearer.
A/O.	3235	Wagon.	68074 Mail.
	68009	Wagon.	

Regular Orders.

R	66227	P. & R.	68076 U. S. Ex.
	66960	P. R. R.	68077 U. S. Ex.
	68075	P. R. R.	

Form No. 34.—COPY OF HECTOGRAPH LIST SENT OUT DAILY.

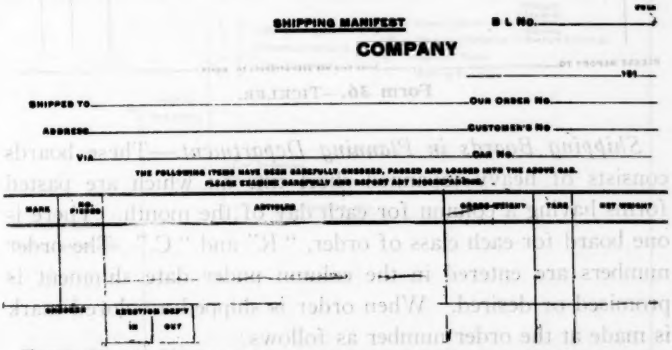
Daily a list, Form 34, is made out of all orders shipped and twelve copies are distributed as follows:

Production clerk	2
Route sheet file	1
D. W	2
D. M	2

Cost department 1
Purchase department 1
Order department 1
Chief draftsman 1

The shop shipping boards are checked by the three-color system.

Copies of B. M., Form 8, are destroyed in all departments on receipt of shipping lists. Form 33 (AP35) is destroyed by the production clerk. Goods are shipped on a shipping manifest, Form 35, the original of which is sent with the bill


SHIPPING MANIFEST
B L No.
COMPANY
SHIPPED TO
OUR ORDER No.
ADDRESS
CUSTOMER'S No
VIA
CAR No.
THE FOLLOWING ITEMS HAVE BEEN CAREFULLY CHECKED, PACKED AND LOADED INTO THE ABOVE CAR.
PLEASE EXAMINE CAREFULLY AND REPORT ANY DISCREPANCIES.
MARK
NO. PIECES
ARTICLES
GROSS WEIGHT
TARE
NET WEIGHT
CHECKED
EXECUTION DEPT
IN
OUT

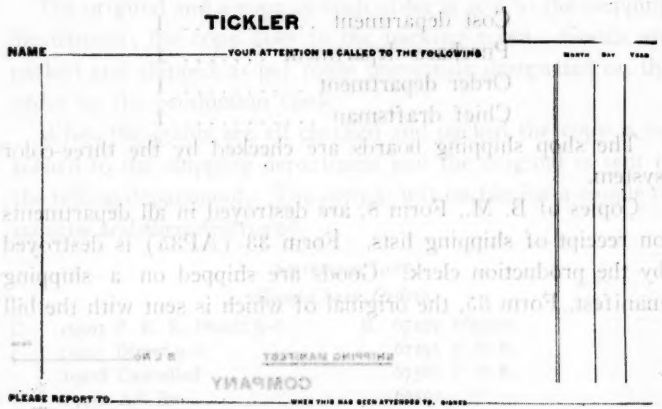
Form 35.—In Triplicate; Original, Duplicate and Triplicate so Marked.

of lading, duplicate filed in the shipping room, and triplicate goes to the cost department. A memorandum is kept in a blank book of job, route, date, etc.

Corresposdence *re* jobs late in shipment is sent to the production clerk for comment.

Tickler.—The tickler, which is now a part of any well regulated shop or office, takes the following form here:

A cabinet with a drawer for each month; in each drawer a large letter file with space for each day of the month. All tickler forms (Form 36) of the superintendent, planning department and shops are sent here for file by dates, and it is the production clerk's duty to daily distribute such ticklers of the day as are on file.



Form 36.—TICKLER.

Shipping Boards in Planning Department.—These boards consists of heavy japanned sheet tins on which are pasted forms having a column for each day of the month. There is one board for each class of order, "R" and "C." The order numbers are entered in the column under date shipment is promised or desired. When order is shipped a colored mark is made at the order number as follows:

If shipped ahead of time, a purple star.

If shipped on time, a blue mark.

If shipped one day late, a red mark.

If shipped two days or more late, a yellow mark.

COST-KEEPING DEPARTMENT.

Analysis of a job order cost.

(Chart III to be read concurrently.)

The grand total selling cost of a job order is made up of (a) total selling cost for this plant and (b) total for other plants. This second item (b) consists of parts or articles manufactured for the job or bought but shipped from other plant direct and not handled at this plant.

(a) Total selling cost (of this plant) consists of (a1) total selling cost of worked materials by divisions. (NOTE: If the job is large, such as a coal-handling device, it is separated into *divisions* such as Hoppers, Springs, Chutes, etc., for convenience of planning, manufacture, etc., and each division is given a letter. If, for instance, the job order were 2035 the divisions would be known as 2035A, 2035B, etc. The accounts of the several divisions are kept separately on all forms until the final cost summary is reached. This is not only a convenience in

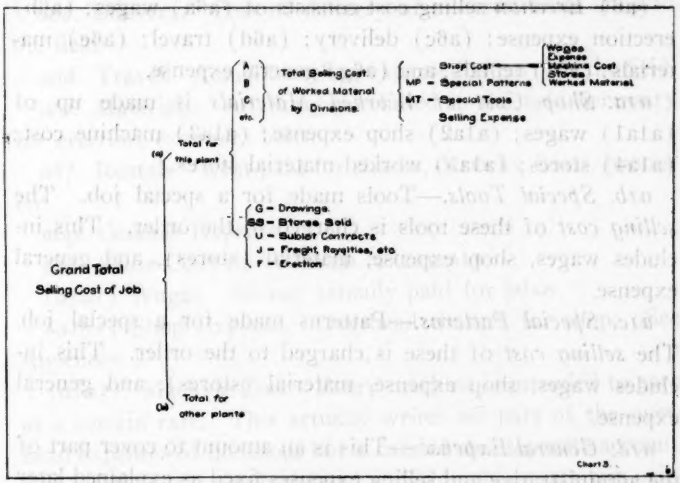


Chart 3.

cost keeping but is essential to estimating and replacing; (a2) Drawings, G; (a3) Stores sold, SS; (a4) Sublet contracts, U; (a5) Freight, royalties, etc., J; and (a6) Erection, F.)

(a1) *Total Selling Cost of Worked Materials by Divisions.* —Each division is made up of (a1a) shop cost of worked material; (a1b) special tools; (a1c) special patterns; and (a1d) general expense.

(a2) *Drawings G.*—The cost of drawings is made up of (a2a) wages; (a2b) drawing expense; (a2c) general expense.

(a3) *Stores Sold* consist of stores delivered with an order

as part of the order but direct from the storeroom and not made up as part of any of the worked material divisions. This cost consists of (a3a) invoice price of stores and (a3b) general expense.

(a4) *Sublet Contracts* are parts of jobs let out and consist of (a4a) contract price and (a4b) general expense.

(a5) *Freight, Royalties, etc.,* are charged directly against the order without any change or additions to it for overhead expense.

(a6) *Erection* selling cost consists of (a6a) wages; (a6b) erection expense; (a6c) delivery; (a6d) travel; (a6e) materials; (a6f) rentals; and (a6g) general expense.

a1a. Shop Cost of Worked Materials is made up of (a1a1) wages; (a1a2) shop expense; (a1a3) machine cost; (a1a4) stores; (a1a5) worked material stores.

a1b. Special Tools.—Tools made for a special job. The *selling cost* of these tools is charged to the order. This includes wages, shop expense, material (stores), and general expense.

a1c. Special Patterns.—Patterns made for a special job. The *selling cost* of these is charged to the order. This includes wages, shop expense, material (stores), and general expense.

a1d. General Expense.—This is an amount to cover part of the administrative and selling expenses fixed as explained later under General Expense.

a2. DRAWINGS, G.

a2a. Wages. Pay actually given to the draughtsmen employed on the drawings of the job.

a2b. Drawing Expense. Part of the overhead charges as explained under *Shop Expense.*

a2c. General Expense. Same remarks as (a1d).

a3. STORES SOLD.

a3a. Invoice Price of Stores, actual buying cost of material to be sold, no shop work to be done on them.

a3b. General Expense. Same remarks as (a1d).

a4. SUBLET CONTRACTS

a4a. Selling Cost of Contract is price at which contract is let.

a4b. General Expense.

a6. ERECTION.

a6a. Wages. Pay to workmen actually in the field.

a6b. Erection Expense. Part of overhead charges, such as superintendent erection department and necessary men retained at all times when not chargeable to any particular order. Repairs to equipment of departments, etc.

a6c. Delivery. Cost of delivery of material from shop to the field.

a6d. Travel. Actual traveling expenses of party in the field.

a6e. Materials. Anything purchased in the field necessary for erection, example: bolts.

a6f. Rentals. Rent paid for shacks, derricks, boats, booms, etc.

a6g. General Expense.

a1a. SHOP COST OF WORKED MATERIALS.

(*a1a1*) Wages. Money actually paid for labor.

(*a1a2*) Shop Expense. Overhead charges for shop. See Appendix I.

(*a1a3*). Machine Cost. Every machine is intended to earn at a certain rate. This actually writes off part of the overhead expenses of the shops but is carried as a separate account. The machines are numbered and a machine-hour rate was originally determined as follows:

[Depreciation (7% annually) + maintenance (supplies and repairs) + small tools + power] ÷ 2500 = Machine-hour cost.

Power is the annual cost of the power actually absorbed by the machine.

Small tools. Was estimated.

The figures in the numerator are for a year. This sum is divided by 2500 on the estimate of 2500 machine hours per year.

The machines are divided into groups, each group containing machines of approximate similarity and power. The aver-

Chart 4.

SHOP NO.	DEPT. NO.	KIND OF MACHINE	SIZE	MAKER	COST PER HOUR	DEPRECIATION	MAINTENANCE	SMALL TOOLS	POWER	REMARKS	HISTORIC	PACING	PIECE WORK
1B	83	VER. BORING MILL	120"	POND MACHINE CO.	52.3	482 20	381 00	164 44	501 30		43	25	34
2B	84	" "	84"	" "		457 10	238 00	164 44	417 75		42	23	30
3B	85	" "	64"	GISHOLT MACHINE CO.		464 04	214 00	164 44	278 50		148	12	20
4B	87	VERT. BORING MILL	37"	BALLARD MACHINE TOOL CO.	25.3	202 00	142 00	98 66	206 80		76	9	15
5B	86	" "	30"	WARNER & SWASEY		139 40	142 00	98 66	206 80		93	9	15
6B	82	HORIZ. "	84"x36"	NEWTON MACHINE CO.		500 10	154 00	98 66	139 25		43	7.5	10
7B	88	" "	18"x18"x36"	BIRNIE MACHINE TOOL CO.		138 00	89 00	98 66	97 40		65	9	7
8B	81	" "		H.M.&M.		93 00	226 00	98 66	194 95		100	9	14
1C	102	GEAR CUTTER	84"	GOULD & EBERHARDT	24.5	345 41	170 00	179 39	276 50		143	10	25
2C	119	" "	72"x6	BROWN & SHARPE		248 10	120 00	179 39	181 06		37	18	13
3C	126	" "	60"x3	" "		178 28	120 00	179 39	153 17		"	"	11
4C	111	" "	48"x2	" "		178 40	100 00	179 39	153 17		"	"	11
5C	117	" "	36"x4	" "		121 80	140 00	179 39	188 30		"	"	9
6C	118	" "	36"x3	" "		83 10	100 00	179 39	111 40		"	"	8
7C	115	" "	36"x2	" "		84 00	100 00	179 39	111 40		"	"	8
8C	120	" "	48"x5	" "		214 54	100 00	179 39	153 10		69	6	11
9C	109	" "	48"x5	" "		210 78	100 00	179 39	153 10		"	"	11
10C	116	" "	36"x3	" "		73 46	100 00	179 39	111 40		37	18	8
11C	107	" "	48"x4	" "		99 84	100 00	179 39	153 10		"	"	11

Chart 4.

age of the rates for any group is taken for the rates of all machines in the group. (Chart IV.)

Obviously for any piece of work machine cost = machine hours × machine-hour cost, machine hours being the actual running time of the machine on the work in question.

(*a1a4*) Stores. Invoice price of stores drawn from the storeroom and used on an order.

(*a1a5*) Worked material. A certain class of stores are made in the shop and a stock kept in the storeroom. This item (a1a5) is the *shop* cost of this class of stores when drawn for use on an order.

METHOD OF COST KEEPING FOR A JOB ORDER.

(Chart V to be read concurrently.)

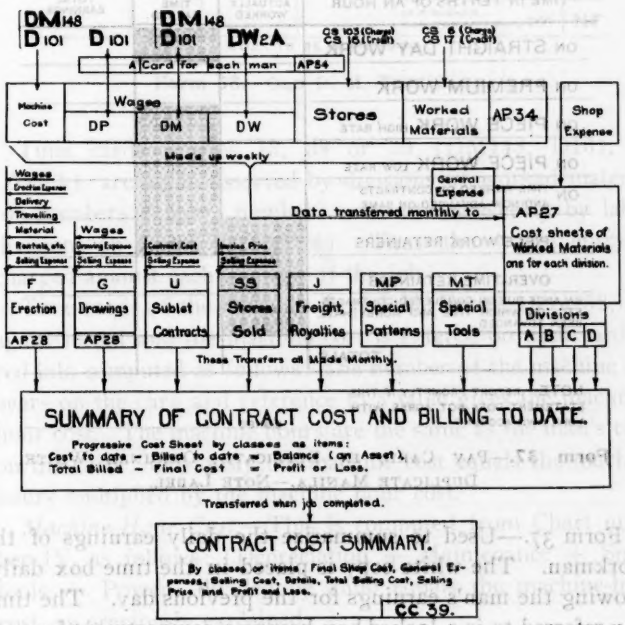


Chart 5.

Time cards of work done are turned in to the cost department from the bulletin board daily. Times of starting and finishing the work are stamped on the cards, and the rate of pay appears on the cards. Two boys compute and enter on the card this elapsed time and the wages due the man. The cards are then sorted by men's numbers. Daily the pay for each man is computed and entered on Form 37 (AP54) in duplicate. The man receives the white copy daily through the time box, showing his earnings for the previous day, the yellow copy being filed for computing the weekly wages due the man.

AP 54

WORKMAN'S NAME..... WORKMAN'S NO. D.....

YOUR TIME AND EARNINGS WERE AS FOLLOWS ON ..... DAY OF WEEK ..... DATE .....

TIME IN TENTHS OF AN HOUR	TIME ACTUALLY WORKED	ADDITIONAL TIME EARNED	TOTAL EARNINGS
ON STRAIGHT DAY WORK			
ON PREMIUM WORK			
ON PIECE WORK HIGH RATE			
ON PIECE WORK LOW RATE			
ON TIME WORKED ON CONTRACTS AND NOW ADVANCED ON SAME			
PIECE WORK RETAINERS			
OVER-TIME RETAINERS			
BALANCE DUE ON COMPLETED CONTRACTS, ETC., ON WHICH TIME AND MONEY HAVE BEEN ADVANCED			
TOTALS			

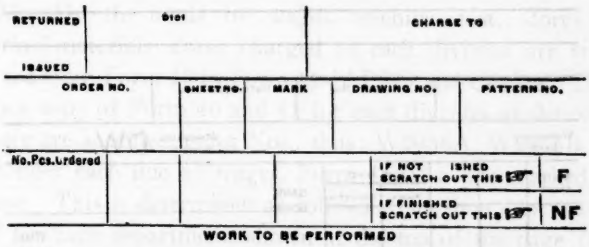
NOTE REPORT PROMPTLY IF THIS STATEMENT DOES NOT AGREE WITH YOUR OWN ACCOUNT

SIGNED.....

Form 37.—Pay Card, in Duplicate; Original White, Duplicate Manila.—Note Label.

Form 37.—Used to summarize the daily earnings of the workman. The white copy is placed in the time box daily, showing the man's earnings for the previous day. The time box referred to is a locked box having a compartment for each

man. This box is opened fifteen minutes before starting time in the morning, and a boy is stationed at the box while open. It is closed at starting time. This box is employed in lieu of a time clock, any men coming after the box is locked have their time stamped at the Order-of-Work Clerk's desk. Only the office use a time clock for arrival and departure. It is worth noting that everybody from the president of the company down stamp the office clock, and the moral effect of this is keenly felt.



Form 38.—Old D. M. Time Card.

Time cards, Forms 13, 38 or 39 (DM148, D101, or DW2A), are again assorted by divisions of worked material job numbers (charge number) and entered weekly by labor divisions on Form 40 (AP34). Thus the wages entry is charged against each division of the job.

Time cards of the machine shop, DM, Forms 13 or 38, are again sorted and the machine cost is entered on each card in red ink, computed as follows: The number of the machine appears on the card and reference to a table gives the machine-hour cost. The machine hours are the same as the man's time on the card. Obviously the machine cost equals the machine hours multiplied by the machine hour cost.

Machine-Hour Cost.—This is computed from Chart number IV, as follows: (Depreciation + Maintenance + Small tools + Power) divided by 2500 equals the machine-hour cost, as previously explained.

IN
OUT

CHARGE SYMBOL

SHIPPING ORDER NUMBER | SHEET NUMBER | MARK | OPERATION NUMBER

DEPARTMENT DW | TOTAL TIME FOR HIGH PRICE | IF JOB IS NOT FINISHED SCRATCH OUT THIS F | F

DAY RATE..... | IF JOB IS FINISHED SCRATCH OUT THIS F | NF

No. of Pieces Ordered	Total Price. High	Total Price. Low	No. of Pieces Finished	Amount Earned	Man's Time	Machine Time	Machine No.

Drawing No. | Pattern No. | Instruction Card |

Workman's Name | Workman's Number DW

Route Sheet	Pay Sheet	Cost Sheet Man's	Cost Sheet Machine's

GANG BOSS.....

DW 12A

TIME CARD

Form 39.—D. W. Time Card for Bonus Work.

BONUS ON PREPARATION | TOTAL BONUS | BONUS PER PIECE

BASE RATE

Time for Preparation	High Rate	Low Rate	Time per Piece	High Rate	Low Rate

REMARKS.....

GANG BOSS

INSTRUCTIONS	Shape of Tool	Cut	Feed	Speed	Time

Form 39.—Reverse. SIGNED.....

After the machine cost is entered on the cards they are again assorted by job divisions and the machine cost is entered weekly on Form 40 (AP34), under column D. C.

The value of stores drawn from the storeroom and applied to the job is entered weekly on Form 40 from slips, Form 12 (CS103), and likewise the value of worked materials drawn from the store and applied to the job is entered on Form 40 from slips, Form 17 (CS6). Before entering values of stores drawn, any credits from Forms 26 (CS16) and 27 (CS8) are deducted.

Monthly the totals of wages, machine cost, stores and worked-materials stores charged to each division are transferred from Form 40 to Form 41 (AP27) and checked. There is one copy of Form 40 and 41 for each division of the order. These are specified as lot Nos., thus: W2035A, W2035B, etc.

Under each line of wages, Form 41, is a line marked expense. This is determined as follows: The shop expense factor for each department, shown at the top of the page (how found and functions shown later), is multiplied into the wages for the month in the corresponding department and this is entered as expense.

AP 34

MONTH	WEEK	WAGES				STORES	WORKED MATERIALS	
		DC	DP	DM	DW			
	1							
	2							
	3							
	4							
	5							
TOTALS								
	1							
	2							
	3							
	4							
	5							
TOTALS								
	1							
	2							
	3							
	4							
	5							
TOTALS								

Form 40.—Weekly Summary of Shop Cost.

DETAIL COST SHEET OF WORKED MATERIALS ON CONTRACT WKD. MTL. LOT No.

DESCRIPTION:

EXPENSE FACTORS USED				DATE STARTED	DATE FINISHED	CHARGE TO
DC	DP	DM	DW			

MONTH													TOTAL
DC	WAGES												
	SHOP EXPENSE												
DP	WAGES												
	EXPENSE												
DM	WAGES												
	EXPENSE												
DW	WAGES												
	EXPENSE												
	WAGES												
	EXPENSE												
STORES													
MISCEL. INCLUDING WORKED MATERIALS													
SHOP COST FOR MONTH													
SHOP COST TO DATE													

GENERAL EXPENSE	
TOTAL SELLING COST	
ESTIMATED SHOP COST	

TOTAL QUANTITY PRODUCED — PIECES / FEET / POUNDS — SHOP COST PER UNIT — PIECE / FOOT / POUND

SUMMARY OF MATERIALS USED	IRON CASTINGS						STEEL SHAPES		STEEL PLATES		MISCEL. STORES
	WEIGHT	PRICE PER POUND	WEIGHT	PRICE PER POUND	WEIGHT	PRICE PER POUND	WEIGHT	PRICE PER POUND	WEIGHT	PRICE PER POUND	AMOUNT

REMARKS:

Form 41.—MONTHLY COST SHEET OF WORKED MATERIALS, ACCOUNTS OF CONTRACT.

Example.—Wages DP $.45. Factor DP .50.

Expense DP is $.45 × .50 = $.23.

This expense covers what is commonly called overhead or shop expense (exclusive of machine cost) as shown in accounts A and D, Appendix 1.

Average Shop Expense Factor.—The average shop expense factor is determined as follows: The total shop expense for a period of three years is apportioned to the several shops in proportion to the amount of this expense incurred by each. This expense in each shop is then divided by the total productive wages in the shop for the same period. In this way a different factor for each shop is obtained, for example:

DP..... 50%
DM..... 110%
DW..... 75%

(NOTE.—These, of course, are not the actual figures at this plant. These factors are applied as shown on Form AP27, and when applied in a normal year should cover all expense incurred under Accounts A and D. (See Appendix 1.) These expenses are such as experiments for the benefit of engineering and manufacturing, heat, light and power, planning connected with machines and materials, etc., and in the several departments DD drawing, DM machine shop, DP pattern shop, and DW wrought shop, such expenses as are connected with manufacturing that cannot be charged to any particular order. A study of this with tables of accounts A and D should make the matter clear.)

The factor as found for the machine shop is not used as it now stands, but is again modified as follows: In a certain class of repeat work (gear cutting for silent gears) one man can run a number of machines. Now, since the factor is applied to wages and since in this special case the ratio of wages to output is greatly reduced, it becomes necessary to increase the factor applied to this special line and decrease that used for the remainder of the shop product in order that this line carry the proportion of shop expense warranted by the tools used, the floor space occupied, and all other considerations governing expense. Therefore the readjustment might be as follows:

DP..... 50%
DW..... 75%
DM.....100%.....Regular
DM.....120%.....Special

This is so adjusted that 50% wages DP + 75% wages DW + 100% wages DM (regular) + 120% wages DM (special) = total shop expense.

After all wages, expenses, machine-cost stores and worked materials are entered in the monthly column of Form 41 (AP27), the total of column for the month obviously gives the shop cost for the month. Theoretically all money spent

on the production, all expenses incidental to production, and the product's share of the shop expenses inherent to all shop output have been charged.

This sheet is progressive in that monthly costs are carried forward and totaled to date at the end of every month, resulting in shop cost to date.

After the *total shop cost* is arrived at, addition of the general expense gives the total selling cost. A general expense factor is used in a manner similar to the shop expense factor. It is determined as follows:

General Expense Factor.—This factor is used to cover what are known as administrative and selling expenses, Account C. These include such items as office salaries, wages and watchmen; repairs and maintenance to all buildings other than shops; planning connected mainly with workmen; contract and sales department; branch officer; shipping room, etc. (See Account C, Appendix 1.)

General expense is distributed over the following items: (1) Shop cost of worked materials; (2) Drawings; (3) Erection; (4) Stores sold; and (5) Sublet contracts.

A factor was first derived by dividing the average general expense for a number of years by the average of the total of the above five items for the same period. For demonstration let us use 30% as the derived factor. Obviously 30% is an excessive amount to saddle on such accounts as erection, stores sold, and sublet contracts. In the case of erection a general expense factor was determined from items covered by Account E (Appendix 1). This account in a general way covers all expenses of erection gangs that cannot be charged to any particular order. This Account E for a number of successive years divided by the *wages* in erection department for the same period gives the erection expense factor.

In the case of stores sold and sublet contracts a factor was determined from good commercial practice. This is arbitrary and depends upon what the product will stand without killing orders.

Having reduced the factor over some of the items, it be-

comes necessary to increase the factors used in the case of drawings and shop cost of worked materials enough to equalize the total general expense of the plant when found by these different factors applied to the shop cost of the respective items, and the total general expense as found if the first derived factor were used throughout all the items.

The factor applied to worked materials is a little larger than that applied to drawings, for several reasons. Drawings are not affected by shipping or stores. Also if the machine hours fall below those used when computing the machine-hours cost, the power that will be written off in machine cost is reduced and this margin is covered by this factor. Also if several machines are run by one man in the DM the total shop cost of the products at these machines are reduced. There is no parallel saving in the drawing department. The total shop cost being reduced it is necessary to increase the factor.

Whereas a factor of 30% was first derived, after all these adjustments the factors would look something like the following:

Worked materials	32%
Drawings	28%
Erection	20%
Stores sold	15%
Sublet contracts	10%

(NOTE.—These are *not* in any proportion to the Company's figures.)

These factors are applied as follows:

(1) Shop cost of worked materials × 1.32 = selling cost of worked materials, Form 41.

(2) Shop cost of drawings × 1.28 = selling cost of drawings.

(3) 1.20 wages + erection expense + deliveries + traveling + material bought in the field + rentals = selling cost of erection contract.

(NOTE.—Factor is applied to wages only. See Form 42 (AP28).

(4) 1.15 invoice price of stores sold = selling cost of S. S., Form 43 (AP58).

AP 39

DETAIL COST SHEET OF DRAWINGS, AND DELIVERY AND ERECTION ON CONTRACT No. _____

DESCRIPTION: _____

DRAWINGS						DATE STARTED		DATE FINISHED		CHARGE TO	
MONTH											TOTAL
WAGES											
DRAW. EXPENSE											
MISCELLANEOUS											
DRAW. ROOM COST FOR MONTH											
DRAW. ROOM COST TO DATE											
									GENERAL EXPENSE		
EXPENSE FACTORS USED									TOTAL SELLING COST		
DRAWING	ERECTION	SELLING							ESTIMATED DRAW. ROOM COST		

DELIVERY AND ERECTION						DATE STARTED		DATE FINISHED		CHARGE TO	
MONTH											TOTAL
WAGES-FA											
ERECTION EXPENSE											
DELIVERY-FF											
TRAVELLING-FH											
MATERIAL-FM											
RENTALS, ETC.-FS											
COST FOR MONTH WITHOUT SELL EXP.											
COST TO DATE WITHOUT SELLING EXPENSE											
									GENERAL EXPENSE		
REMARKS:									TOTAL SELLING COST		
									ESTIMATED COST WITHOUT SELLING EXPENSE		

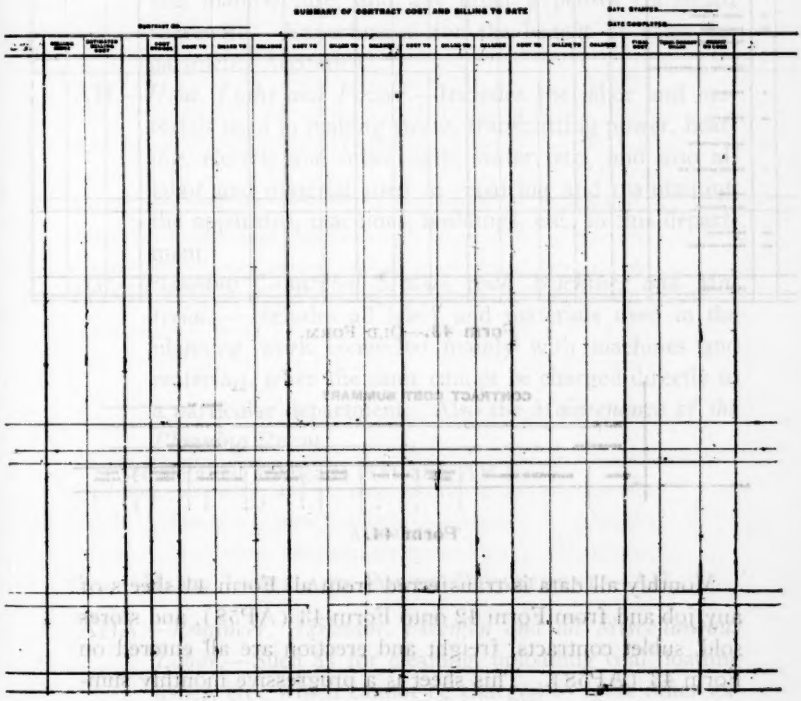
Form 42.—MONTHLY COST SHEET FOR DRAWING AND ERECTION ACCOUNTS OF CONTRACT.

(5) 1.10 contract price of sublet contracts = selling cost of sublet contracts, Form 43 (AP58).

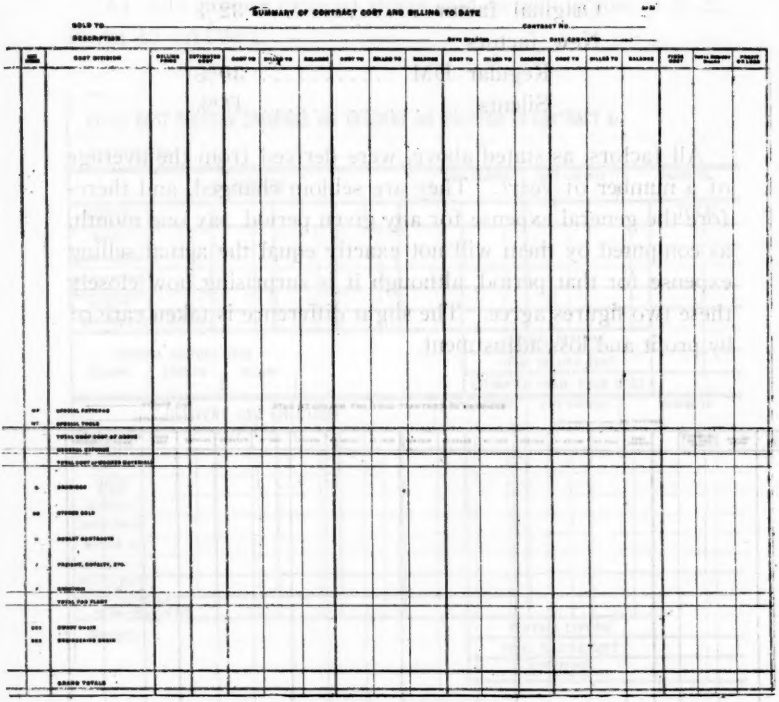
There is still another correction to be made in the case of this particular shop before the factors are settled. In the case of a special line of goods called Silent Orders (silent gears and chain) it is found that a far greater expense is incurred per dollar of order to get orders in this special line than is incurred getting the regular orders. So the factor on worked materials is again altered, and since the value of silent orders is but a small proportion of the whole shop output the factor on this line rises rapidly thus:

Original factor..... 32%
New factors:
Regular DM 30%
Silents 45%

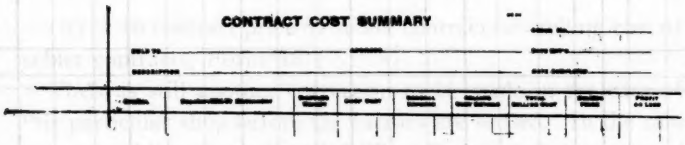
All factors, as stated above, were derived from the average of a number of years. They are seldom changed, and therefore the general expense for any given period, say one month, as computed by them will not exactly equal the actual selling expense for that period, although it is surprising how closely these two figures agree. The slight difference is taken care of by profit and loss adjustment.



Form 43.—NEW FORM.



Form 43.—OLD FORM.



Form 44.

Monthly all data is transferred from all Form 41 sheets of any job and from Form 42 onto Form 43 (AP58), and stores sold, sublet contracts, freight and erection are all entered on Form 43 (AP58). This sheet is a progressive monthly summary of details. All goods billed to customer during the progress of the work are charged off this sheet by the pound at a

pound price, and the difference between the selling cost to date and the goods billed to date = balance, which is an asset.

When a job is completed.—All items are transferred from Form 43 to Form 44 (CC39), which is a *final* contract cost summary of the job by divisions, showing the profit or loss on same.

APPENDIX.

ACCOUNT A.—AUXILIARY DEPARTMENTS.

AE.—*Experiments for the Benefit of Engineering and Manufacturing.*—Includes all labor and material used in the making of experiments for the benefit of engineering and manufacture, and any other expenses connected therewith. Experiments for the benefit of sales department, Account C.

AH.—*Heat, Light and Power.*—Includes the labor and materials used in making steam, transmitting power, heating, electric and other light, water, etc., and also all labor and material used in repairing and maintaining the apparatus, machines, buildings, etc., in this department.

AP.—*Planning Connected Mainly with Machines and Materials.*—Includes all labor and materials used in the planning work connected mainly with machines and materials, when the same cannot be charged directly to a particular department. Also the *Maintenance of the Planning Room.*

(See detailed subheads AH and AP.)

AUXILIARY.

HEAT, LIGHT AND POWER DEPARTMENT.—AH.

AHA.—*Engineer, Assistant, Firemen and all Miscellaneous Labor.*—Such as for cleaning, unloading coal, loading ashes, etc., which cannot be charged to some other expense symbol of this department.

AHB.—*Fixture and Furniture.*—Includes such items as benches, cupboards, shelving, partitions, desks, clocks, etc., *repairs and maintenance but not new.* Charges against new or improvements charged against Account X.

Stores and supplies, including stationery, which cannot be charged to one of the other expense items of this department. Also all small tools used in this department that are not borrowed from the tool room. *Repairs, Maintenance and New.*

AHC.—*Boilers, Stacks, Flues and Ashes Elevators.*—Includes all labor and material for overhauling, scaling, testing, etc., *repairs and maintenance but not new.* Charges for new and improvements to Account Y.

AHP.—*Power and Transmission of Power.*—Includes engines, generators, motors, compressors, pumps and piping to main distribution valve, and all wiring to main distributing switch, *repairs and maintenance but not new.* New and improvements charged to Account Y.

AHZ.—*Buildings, Repairs of.*—Includes *repairs and maintenance of* buildings, floors, steam heat and water-supply system from main valves, including all alterations and repairs to patterns and drawings required by this work. Charged for new and improvements charged to Account Z.

PLANNING FOR MACHINES AND MATERIAL.—AP.

APA.—*Shop Engineer, Production Clerk,* and also all clerks, messengers, etc., whose work is mainly connected with *planning for machines and materials,* and which is not chargeable to any other department order.

APB.—*Fixtures, Furniture and Apparatus,* includes such items as benches, shelving, partitions, desks, tables, stools, chairs, clocks, electric fans and other electrical apparatus, wiring, etc. *Repairs and maintenance but not new.* New and improvement charged to Account X.

Stores and supplies, including stationery, used in the planning room and which cannot be charged to a particular manufacturing, equipment, construction or other department order.

APR.—*Reclamation for Errors* made in planning for machines and materials.

SYNOPSIS OF GENERAL EXPENSE.—ACCOUNT C.

CA.—Accounting department.
CB.—Purchasing department.
CC.—Sales department.
CD.—Estimating department.
CE.—Stenographic department.
CF.—Photographic department.
CG.—Order department.
CH.—Entertainment department.
CJ.—Replacements and diplomatic extra work.
CK.—Exchange and collection expense.
CL.—St. Louis office.
CM.—Seattle office.
CN.—New York office.
CO.—Silent chain sales department.
CP.—Pittsburgh office.
CR.—Lunch room.
CS.—Stores department.
CU.—Boston office.
CV.—Buffalo office.
CW.—Shipping room.
CX.—Charities and contributions.
CY.—Telephones and telegraphs.
CZ.—Repairs to office building.

MANUFACTURING DEPARTMENT.—ACCOUNT D.

DD.—Drawing room.
DM.—Machine shop.
DP.—Pattern shop.
DW.—Wrought shop.

MACHINE SHOP DEPARTMENT.—DM.

NOTE.—All other shop departments are made up of similar subdivisions, so only the machine shop is exhibited.

DMA.—*All Functional Foremen, their Assistants* and all labor that cannot be charged to a particular manufacturing, construction, equipment, or other department order, or to one of the other expense items of this department, including machine-shop tool room. Also the rate setters and route clerks in the planning department that work only for the benefit of the machine shop.

DMB.—*Fixtures and Furniture, Including Tool Room.*—Includes such items as benches, cupboards, shelving, racks, partitions, shop lockers, desks, tables, stools, clocks, tote-boxes, trucks, scales, bulletin boards, etc. *Repairs and maintenance but not new.* Charge for new to Account X.

Stores and Supplies, including stationery; such as waste, oil, etc., which cannot be charged to a particular manufacturing, equipment, construction or other department order; or to some one of the particular expense items of this department.

DME.—*Small Tools.*—Includes the repairing and maintaining of small tools such as drills, chisels, planer and lathe tools, dies, milling cutters, etc. Also the making of these tools, including the tool steel used therein, machining and dressing, and all tools of this kind that are drawn from stores for the purpose of only maintaining the tool-room stock and not with a view of increasing same. Charges for new tools *increasing the tool-room stock* charged to Account X.

Large Hand Tools.—Includes those tools which do not depreciate rapidly by use, and which add to the permanent value of the shop equipment, such as vises, anvils, pulleys, blocks, jacks, etc. *Repairs and maintenance* but not new. New and improvements charged to Account X.

Grinding and Dressing of Tools.—Includes the grinding and dressing of old tools, and the treatment of new and old tools. The dressing of new tools is not included in this item, the work being charged to the order for the manufacture of tools.

DMH.—*Special Tools.*—Includes such tools as jigs, gages and templates which cannot be charged to a particular manufacturing, equipment, construction or other department order. *Repairs and maintenance and new.*

DMK.—*Work on Defective Material.—i. e.,* work done by the shops on material that has concealed defects, that is, such as disclose themselves only after some work has been done.

NOTE.—Work done on defective material, the defect in which could and should have been discovered before any work is done, is to be charged against CSR, it being due to an error on the part of the stores inspector.

DMM.—*Machines Operated by Hand or Power, Including Cranes.*—Includes all machines with their countershafts or motors and all cranes operated by hand or power. *Repairs and maintenance but not new,* including any alterations and repairs of patterns and drawings required by this work. *New* and improvements charged to Account Y. Also,

Power and Transmission of Power.—Includes all compressed air lines within the buildings (main), line shafting and their motors and all electrical fixtures and wiring for both power and light. *Repairs and maintenance but not new.* Charges for new and improvements except electrical fixtures and wiring charged to Y. Fixtures and wiring charged to Account X. Also,

Belting.—Includes labor and materials used for repairing and maintaining all belts in the machine shop. Also the cost of new belting replacing old, and all belt lacing, glue and small tools used by the belt fixer. *New* belts for new machines and for line shafting charged to different divisions of Account Y.

DMN.—*Retainers.*—Both for piece work (DMNP) and overtime (DMNT) connected with worked materials, and construction and part construction.

DMR.—*Reclamation for Errors* made by the machine shop or tool room.

DMV.—*Repairs to Patterns.*—Includes all labor and materials used in repairing patterns, and in handling and transporting patterns, including freight and expressage. Also *repairs and maintenance of pattern shed.* Charges for *new patterns* that cannot be made to a particular manufacturing, equipment, construction or other department order, are to be made to Account X.

DMW.—*Painting and Moving, and Other Miscellaneous Labor.*—Including all labor and material used in painting and oiling worked material produced in the machine shop, and also all labor used in moving materials in the machine shop. Also, oiling and cleaning of all line shafting and motors, machines when not in operation, shop floors, windows, etc.

DMZ.—*Buildings.*—Includes *repairs and maintenance* of buildings, washrooms, washroom fixtures, urinals, waterclosets and soil pipes, inside of machine building, exclusive of that part occupied by offices. Also, steam-heat and water-supply system from main valves, including any alterations or repairs to patterns and drawing required by this work. Charges for new and improvements are to be charged to Account Z.

ERECTING EXPENSE.

ERECTION DEPARTMENT.—E.

When any of the following symbols are used, they indicate that some one of the following kinds of expense has been incurred chargeable to the above department.

EA.—*Superintendent of Erection Department, Assistants, Foreman Gang Bosses, Clerks and Workmen* when their work cannot be charged to a particular *delivery* or *erection* order.

EB.—*Fixtures, Furniture, etc.*—Includes benches, cupboards, shelving, partitions, desk and office furniture, clocks, tool boxes, etc. *Repairs and maintenance but not new.* Charges for new or improvements are to be made on proper written authority to (XCB). Also,

Stores and Supplies, Including Stationery, which cannot be charged to a particular manufacturing, equipment, construction or other department order, or to one of the other expense items of this department.

EE.—*Small Tools.*—Includes all small tools purchased or manufactured, which are used by hand and must be replaced frequently on account of wear, and hence can not be considered as adding to the permanent value of the equipment, such as hammers, wrenches, chisels, rope, bars, shovels, etc., including repairs and maintenance of same. Also,

Large Hand Tools and Machines.—Includes those tools which do not depreciate rapidly from use and wear, and which add to the permanent value of the equipment, such as hoisting engines, winches, etc. *Repairs and maintenance but not new.* Charges for new and improvements are to be made on proper written authority to (XEF).

EM.—*Miscellaneous.*—Includes all cost of living, traveling, and stores and supplies chargeable to this department and which cannot be charged to a particular *delivery* and *erection* order.

ER.—*Reclamation for Errors* made by this department.

EW.—*Starting up.*—Includes all extra expense incurred in starting up machinery, which cannot be charged to any particular *delivery* and *erection* order.

EU.—*Unloading and Loading.*—Includes that which cannot be charged to any particular *delivery* and *erection* or-

der. This refers especially to handling of erection tools.

EZ.—*Buildings, Repairs of.*—Includes repairs and maintenance of any permanent building that might be built by this department. Charges against new or improvements are to be made on proper written authority to (ZE).

Account X referred to in the foregoing is part construction, and includes fixtures, apparatus, etc., part of the cost of which goes to construction and part to shop expense or general expense and which increases the permanent value of the plant.

Account Y is construction, and consists of machinery, tools and motive power that adds to the permanent value of the plant.

Account Z is construction, and consists of *real estate,* buildings, etc.

A COMPARISON OF THE COST OF DREADNOUGHTS IN ENGLAND, GERMANY, FRANCE, AUSTRIA AND THE UNITED STATES.

Translated from an article of Naval Constructor LOUIS BARBERIS, R. I. N., in "Rivista Marittima," by A. CONTI, Associate.

One often reads in technical magazines and official publications comparisons on the cost of foreign Dreadnoughts. Such figures are usually published without comment, as if they were directly comparable, but, unfortunately, as every Navy has a different system of accounting, great care is required in making comparisons. Given the cost of a certain ship, it is first necessary to know whether it includes the price of service and spare ammunition, torpedoes and spare guns.

In some Navy budgets, Austrian-Hungarian and German, for instance, when the cost of a ship is stated it means the ship complete, including torpedoes as well as service and spare ammunition. The British Navy budgets, however, contain separate items for the purchase of ammunition and torpedoes, both for new ships and ships already in commission; therefore such material is not included in the cost charged to a new ship. The French estimates of the cost of a new ship include only the service ammunition.

In the official documents of the United States (The Acts of Congress, for instance), in fixing the cost of a new ship it is understood that it has reference only to the hull and machinery, exclusive of armor, armament, anchors and chains, torpedoes and ammunition.

GENERAL EXPENSES.

Another point on which the bookkeeping of the various navies differs a great deal is the allotment of general expenses. When a ship is bought from a private shipyard, its tender includes not only all the overhead and operating expenses,

but covers insurance, taxes, cost of trials, etc. Besides these expenses which, from the taxpayer's viewpoint, should be charged to the cost of new ships, there are others that never appear in the navy estimates. These are the salaries of the designing staffs and of those employed at the experimental tanks and technical laboratories, of the inspectors at the works of contractors and subcontractors, etc., and in some navies where it is customary to detail a large number of officers and men to the ship long before its completion, this expense should also be added. For the allotment of these expenses and of many more that have not been mentioned, the different navies follow the most varied systems.

The British Navy seems on this account the most particular, judging from the minute subdivision of these expenses in the "Dockyard expense account." But if one analyze how these are charged to the new ships building, it will be seen that the allotments are entirely inadequate. During the 1909-10 budgets, for instance, out of £3,021,923 of general expenses only £774,214 was "apportioned and carried to balance sheet," of which £356,883 was charged to new constructions and £417,331 to repairs and upkeep, while the balance of £2,247,709 remained "unappropriated."

If one considers that for the same period the cost of new construction amounted to £9,597,551 and for repairs and upkeep £3,412,733, or a direct total expenditure of £13,010,284, it will be seen that the amount "unappropriated" is hardly negligible.

But this should not be thought the cause of the relatively low cost of English ships, as other navies follow quite the same system, and, if anything, a smaller fraction of the "general expenses" is considered. For the French and American navies we are on this point very well informed. We have for France the interesting report published by the "Comité d'Examen des Comptes des Travaux de la Marine," and for the United States the "Instructions Relative to Accounting at Navy Yards." It appears that for both navies the general expenses do not include salaries or pensions of the dockyard

staffs, nor the interest or depreciation of the navy-yard plants. They include only the salaried help whose work is directly charged to new construction and the cost of material for the general service of the navy yards.

Obviously, then, it is quite impossible to compare these items, The fact that the *Bellerophon* is credited with $546,000 of general expenses, while the same item was estimated at $360,960 for the *Jean Bart* and $544,314 for the *New York,* does not necessarily imply that in France money is better administered than in England or in the United States. Such differences must be attributed solely to the meaning which this item "general expenses" assumes in the various navies.

In Austria the salaries of the navy yard employés do not appear in the budget as chargeable to new construction, so that for ships built in dockyards (the *Admiral Spaun,* for example) the supposition is made that only transient help is employed. In the German budget of 1911-12 there appears the conspicuous sum of 36,719,792 marks under the title "Upkeep of fleet and dockyards," and a large part of this sum is itemized for labor and material that undoubtedly go to the building of new ships. So that for both the German and Austrian navies the "general expenses" inherent in new construction do not explicitly appear in any official document.

Before reviewing the figures gathered from official publications it must be noted that the cost per ton is based on the so-called normal displacement. And this particularly applies to the Navy of the United States, in which, after the Newport conference, by "normal displacement" is meant that with two-thirds (about 2,000 tons) of the maximum amount of fuel. Were not the displacements of American ships decreased by 1,000 tons their cost per ton could not fairly be compared with foreign ships.

COMPARISON OF COST, NAVY YARD AND PRIVATE SHIPYARD.

Before comparing the cost of Dreadnoughts of different navies, a comparison which we will restrict to England, Germany, France, Austria and the United States, we must men-

tion the difference in cost between government and private-built ships. For Germany this is quickly done, as the German estimates do not allow for any differences. For Austria the *Habsburg, Erzherzog, Radetzky* and *Viribus Unitis* were all built or are building in private yards, so that a just comparison is not possible. There remains then England, France and the United States. Officially, in France, government-built ships cost 4 or 5 million francs less than those built by private yards, except in the case of the *Jean Bart* class, where this difference is only one million. In the United States things are reversed, and in rather extraordinary proportion. The *Florida* cost $2,300,000 more than the *Utah*. The cost of the *New York* is estimated at $1,540,000 more than that of the *Texas*, but had it not been for the eight-hour clause the difference for these two ships would probably have been from $2,300,000 to $2,500,000. In England private-built ships cost ordinarily from $200,000 to $400,000 less than government-built ships, but there are examples to the contrary like the *Indefatigable*, which, built in a navy yard and of larger size than the *Indomitable*, cost about $1,000,000 less. These differences have led in the various nations to very animated discussions, and of particular interest are those originated in March, 1911, in the French Parliament and the United States House of Representatives. In the former, during the meetings of March 9, 1911, representative Chaumel remarked that even for the so-called government-built vessels, five-sixths of the parts are supplied by private contractors. Thus for a ship costing about 13 millions, at least 10 millions belong to the machinery, armor and armament that are bought directly from manufacturers. So that it is logical to exclude from this comparison those parts which are purchased, whether the ship be built at a dockyard or by a private concern. Now, on the remainder of work, valued at about 3 millions, can there be an increase of about 33 per cent. if the ship is built by a private yard?

This difference arises only from the fact that the State does not compute salaries or the pensions of the dockyard staffs, nor the material or sinking fund of the capital invested. It is

probable that for the taxpayer there is no sensible difference, as these general expenses not charged to the cost of a new ship amount to just about this difference in cost. On the other hand, the dividend to the stockholders will compensate in part the lower efficiency of the State organization, and the difference either way cannot be substantial. For the United States the opposite phenomenon exists. It must be noted, however, that the American system of bookkeeping in the estimate for a ship to be built at a navy yard allows about $600,000 for general expenses, whereas France allows only about $400,000. Moreover, the American navy-yard laborer enjoys each year a two weeks' vacation with full pay; plus 5 legal holidays also with full pay, involving a yearly loss for the State of $1,664,868, of which perhaps one-third goes to increase the cost of new vessels constructed in navy yards.

In England these differences are so small that they further demonstrate our contention that if everything is taken into account a new ship costs just as much to the taxpayer whether built by the State or by a private concern. The cost of the *Inflexible* class, which was more than that of the *Indefatigable*, must be considered as a special occurrence due to the development of a new type of vessel with extremely powerful machinery. The three *Inflexibles* were given to private concerns, without any competition, a procedure deeply regretted by the "Committee on Public Accounts," and which may in part explain the unusual occurrence.

COST PER TON.

With this preliminary discussion, if one now wishes to compare the cost per ton for ships of various navies, a careful inspection of the official data recorded in Table I is quite necessary. And this inspection, no matter how careful, can give only approximate results, owing to lack of information.

The best way is to consider only private-built ships, subtracting from their official cost the item "general expenses" and adding to it all other expenses borne by navy yards for parts and equipment directly supplied by the Navy.

Table II records this cost subdivided as follows:

1st. Hull complete with equipment;
2d. Propelling machinery;
3d. Armor;
4th. Ordnance.

Under the first heading are included torpedo nets, torpedoes, air compressors, electrical and wireless equipment; all the ships' auxiliaries including motors, all equipment including galleys and mess rooms; it also includes the cost of labor for installing armor and the cost of armor bolts, but all fixed parts pertaining to turrets, such as rollers and guides, turret-turning gear, ammunition hoists, etc., are excluded. This item includes also the armored decks as being part of the hull proper, as well as all permanent fixtures of storerooms and magazines, fixed-ammunition hoists for secondary battery and all ship fittings, including anchors and chains.

The first heading includes also all taxes, insurance, etc., and that such items are not to be neglected is shown by the fact that in France, for instance, they amount to about 900,000 francs for a ship of the *Voltaire* class; the insurance alone amounting to 1.5 per cent. per annum of the value of all parts furnished during the entire period of construction.

Where foreign material is imported and pays duty (in Austria, for instance, British plates and shapes are taxed 30 crowns per ton), the customs duties are added to the cost of the hull.

Heading No. 2 includes the main propelling and auxiliary machinery and their piping, as well as the cost of dock and sea trials.

Under the third heading are included all armor except protective decks, as well as the cost of trial armor plates.

The fourth heading includes main and secondary batteries complete with mountings, and all order-transmitting and fire-control appliances, but does not include any spare or service ammunition, nor any of the permanent fixtures in ammunition magazines, which, as already stated, are included in the first heading.

The relative value of currencies has been taken as follows:

1 Pound sterling	equals $4.85
1 Mark,	.24
1 Crown,	.202
1 Franc,	.193

From table II the following average prices per ton of displacement have been derived omitting armored cruisers:

	Hull and machinery.	Ship complete without ammunition.
England	$182	$401
Germany	214	416
United States	212	417.5
Austria	256	490
France	247	525

DEDUCTIONS FROM THE TABLES.

In regard to cost of production, the nations may be grouped as follows:—England, Germany and the United States, where ships are comparatively cheap, and Austria and France, where ships cost much more.

In the first group, England pays very little for hull and machinery while armor and ordnance are comparatively expensive. In the United States armor is cheap; in Germany this is so with ordnance, although this is probably due rather to the small proportion of the displacement assigned to the battery than to the low cost of German ordnance.

If the *Texas* had not been included in the average the mean cost for the United States would have been slightly less than for Germany. It must be noted that the low cost of the *Nassau* class is for ships with reciprocating engines and 11-inch guns.

In France ordnance is comparatively expensive. This is easily explained for the *Dantons,* owing to the large number of turrets, but is not so justified for the *Jean Barts.* For Austria armor is rather moderate. Due to the fact that her

Table Ia.—ENGLAND.

	Battleships.									
	Dreadnought	*Bellerophon*	*Temeraire*	*Superb*	*St. Vincent*	*Collingwood*	*Vanguard*	*Neptune*	*Hercules*	*Colossus*
Keel laid	Oct., 1905	Dec., 1906	Dec., 1906	Feb., 1907	Jan., 1908	Feb., 1908	Apr., 1908	Jan., 1909	July, 1909	July, 1909
Where built	Navyyard.	Navyyard.	Navyyard.	Shipyard.	Navyyard.	Navyyard.	Shipyard.	Navyyard.	Shipyard.	Shipyard.
Cost, in pounds sterling:										
Hull complete with armor	878,841	832,514	832,046	800,294	800,705	804,725	745,777	829,760	851,066	848,718
Power boats	7,898	4,076	4,344	4,435	5,185	4,110	4,710	5,867	4,210	4,240
Auxiliary machinery	20,009	29,023	27,808	5,707	23,911	23,911	} 251,205	{ 17,023	} 237,968	252,880
Main propelling machinery	296,513	294,719	278,084	281,151	267,848	253,458		{ 233,040		
Gun mounting	381,597	370,697	379,218	417,289	407,654	406,934	427,018	394,776	405,076	402,065
Guns	113,200	116,300	116,300	116,300	142,400	142,400	142,400	141,000	131,500	131,500
Torpedo tubes	7,882	5,018	4,288	4,985	3,965	3,900	4,053	3,860	5,299	5,318
General expenses	111,160	113,339	101,867	32,423	102,848	92,212	32,618	89,932	26,791	26,913
Total	1,813,100	1,765,342	1,743,955	1,662,584	1,754,615	1,731,640	1,607,781	1,715,258	1,661,911	1,671,934
Displacement, tons	18,200	18,900	18,900	18,900	19,600	19,600	19,600	20,200	20,200	20,200

	Battleships.				Battleship Cruisers.				
	Orion.	*Thunderer.*	*Conqueror.*	*Monarch.*	*Invincible.*	*Inflexible.*	*Indomitable.*	*Indefatigable.*	*Lion.*
Keel laid	Nov., 1909	Apr., 1910	Apr., 1910	Apr., 1910	Apr., 1906	Feb., 1906	Mar., 1906	Feb., 1909	Nov., 1909
Where built	Navyyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Navyyard.	Navyyard.
Cost, in pounds sterling:									
Hull complete with armor	962,657	1,019,005	1,043,024	1,045,887	795,901	798,745	807,936	{ 649,052	941,083
Power boats	5,800	4,031	4,250	4,161				{ 4,090	6,085
Auxiliary machinery	20,417	} 256,922	253,239	235,095	484,772	467,018	476,485	{ 19,787	19,938
Main propelling machinery	244,663							{ 381,008	470,233
Gun mounting	432,624	433,625	392,412	429,294	355,901	330,526	345,341	296,417	326,903
Guns	150,000(?)	150,000(?)	150,000(?)	150,000(?)	90,000	90,000	90,000	98,500	120,000
Torpedo tubes	5,350	5,059	5,345	5,059	3,398	8,403	7,713	2,821	4,970
General expenses	98,383	27,186	27,039	27,544	34,023	33,537	33,605	85,094	106,032
Total	1,919,894	1,895,828	1,875,309	1,897,040	1,763,995	1,728,289	1,761,080	1,536,769	1,995,244
Displacement, tons	22,900	22,900	22,900	22,900	17,500	17,500	17,500	19,000	26,800

Table Ib.—GERMANY.

Table Ib.—GERMANY.

	Battleship.							Battleship Cruisers.	
	Nassau.	*Westfalen.*	*Rheinland.*	*Posen.*	*Helgoland.*	*Ostfriesland.*	*Thüringen.*	*Von der Tann*	*Moltke.*
Keel laid	1907	1907	1908	1908	1909	1909	1909	1908	1909
Where built	Navyyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Navyyard.	Shipyard.	Shipyard.	Shipyard.
Cost, in marks:									
Hull, armor, and machinery..	22,270,000	22,270,000	22,270,000	22,270,000	26,700,000	26,700,000	26,700,000	26,000,000	29,150,000
Ordnance and ammunition ...	13,500,000	13,500,000	13,500,000	13,500,000	19,000,000	19,000,000	19,000,000	10,000,000	14,000,000
Torpedo equipment..	990,000	990,000	900,000	990,000	1,400,000	1,400,000	1,400,000	560,000	930,000
Total	36,760,000	36,760,000	36,760,000	36,760,000	47,100,000	47,100,000	47,100,000	36,560,000	44,080,000
Displacement, tons.....	18,500	18,500	18,500	18,500	21,650	21,650	21,650	19,000	23,200

UNITED STATES.

	Michigan.	*South Carolina.*	*Delaware.*	*North Dakota.*	*Utah.*	*Florida.*	*Arkansas*	*Wyoming*	*Texas.*	*New York.*	*Nevada.*	*Oklahoma.*
Keel laid	1906	1906	1907	1907	1908	1908	1909	1909	1910	1911	1911	1911
Where built	Shipyard	Shipyard	Shipyard	Shipyard	Shipyard	Navyy'd.	Shipyard	Shipyard	Shipyard	Navyy'd.	Shipyard	Shipyard
Cost, in dollars:												
Maximum authorized cost..	4,400,000	4,400,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000
Contract price	3,585,000	3,540,000	3,987,000	4,377,000	3,946,000	6,400,000	4,450,000	4,675,000	5,830,000	7,500,000	5,895,000	5,926,000
Actual price paid to contractors	3,647,550	3,657,288	3,921,102	4,412,359	Not completed.							
Expenditures at navyyards..	179,676	202,955	256,371	232,259	Not completed.							
Contract price for armor.....	1,266,067	1,266,067	1,954,474	1,954,474	1,985,152	1,985,152	2,746,628	2,746,628	2,810,700	2,810,000	...	...
Estimate for ordnance and torpedo equipment.....	?	?	?	?	2,250,000	2,250,000	2,540,980	2,540,980	?	?	?	?
Anchors and chains.....	Probable cost $25,000 to $30,000 should be added to above figures.											
Service ammunition.....	?	?	?	?	611,000	611,000	?	?	1,009,500	1,009,500	?	?
Displacement, tons.....	16,300	16,300	20,300	20,300	21,200	21,000	25,400	25,400	26,400	26,400	?	?

Table Ic.—FRANCE.

Name of ship.	*Danton.*	*Mirabeau.*	*Voltaire.*	*Diderot.*	*Condorcet.*	*Vergniaud.*	*Jean Bart.*	*Courbet.*	*France.*	*Paris.*
Keel laid	1906	1906	1906	1906	1906	1906	1910	1910	1911	1911
Where built	Navyyard.	Navyyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Navyyard.	Navyyard.	Shipyard.	Shipyard.
Cost, in francs.										
Department of Naval Construction:										
Cost of ship	.....	.....	43,941,747	43,847,297	43,781,740	43,793,888	.....	.....	50,680,000	50,680,000
D. of N. C., labor, actual	6,300,000	5,800,000	280,000	280,000	280,000	280,000	7,400,000	7,400,000	300,000	300,00
material, actual	31,864,790	32,570,897	570,800	570,800	570,800	570,800	42,363,600	42,363,600	1,428,000	1,428,000
torpedoes	79,200	79,200	79,200	79,200	79,200	79,200	158,400	158,400	220,000	220,000
labor, overhead charges	819,000	754,000	36,400	36,400	36,400	36,400	889,000	889,000	39,000	39,000
material, overhead charges.	945,000	870,000	42,000	42,000	42,000	42,000	889,000	889,000	45,000	45,000
Ordnance, labor, actual	250,000	250,000	250,000	250,000	250,000	250,000	300,000	300,000	300,000	300,000
material, actual	10,289,232	9,958,244	9,958,244	9,958,244	9,958,244	9,958,244	13,000,000	13,000,000	13,000,000	13,000,000
labor, overhead charges.	40,000	40,000	40,000	40,000	40,000	40,000	54,000	54,000	48,000	48,000
material, " "	32,500	53,000	32,500	32,500	32,500	32,500	45,000	45,000	39,000	39,000
Total	50,619,812	50,354,841	55,230,891	55,137,441	55,070,884	55,083,032	65,099,000	65,099,000	66,099,000	66,099,000
Spare ammunition	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	6,000,000	7,623,000	7,623,000	7,623,000	7,623,000
Spare guns	286,333	286,333	286,333	286,333	286,333	286,333	1,110,000	1,110,000	1,110,000	1,110,000
Grand total	56,906,145	56,641.174	61,517.224	61,423,774	61,357,217	61,369.315	73,832,000	73,832,000	74,832,000	74,832,000
Displacement, tons	18,300	18,300	18,300	18,300	18,300	18,300	23,500	23,500	23,500	23,500

Table Id.—AUSTRIA-HUNGARY.

Table 1d.—AUSTRIA-HUNGARY.

	Viribus Unitis.	*Ersatz-Erzherzog-Friedrich.*	*Ersatz-Dandolo.*	*Ersatz-Saida.*
Keel laid	1910	1910	1911	1911
Where built	Shipyard.	Shipyard.	Shipyard.	Shipyard.
Cost, in crowns:				
Hull and machinery	37,000,000	37,000,000	37,000,000	37,000.000
Ordnance and ammunition	23,600,000	23,600,000	23,600,000	25,600,000
Total	60,600,000	60,600,000	60,600,000	60,600,000
Displacement, tons	20,500	20,500	20,500	20,500

MISCELLANEOUS NATIONS.

	RUSSIA.				SPAIN.		
	Gangut.	*Poltava.*	*Sebastopel.*	*Petro-pavlovsk.*	*Espana.*	*Alfonzo XIII.*	*Jaime I.*
Keel laid	1909	1906	1909	1909	1909	1909	1909
Where built	Navyyard.	Navyyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.
Cost	27,225,435 rubles.	27,225,435 rubles.	27,225,435 rubles.	27,225,435 rubles.	45,000,000 pesetas.	45,000,000 pesetas.	45,000,000 pesetas.
Displacement, tons	23,300	23,300	23,300	23,300	15,500	13,500	15,500

	BRAZIL.			ARGENTINA.		AUSTRALIA & NEW ZEALAND.	
	Minas Geraes.	*Sao Paulo.*	*Rio Janiero.*	*Moreno.*	*Rivadavia.*	*New Zealand.*	*Australia.*
Keel laid	1008	1909	1910	1910	1910	1910	1910
Where built	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.	Shipyard.
Cost, in pounds sterling	1,821,400	1,821.400	?	2,190,000	2,190.000	1,800,000	1,800,000
Displacement, tons	19,500	19,300	32,000	27,200	27,200	18,800	18,800

Table II.—Partial and Total Cost, in Dollars, Corrected, of Private-Built Ships.

Name of ship.	Hull and equipment.	Propelling machinery.	Armor.	Ordnance without ammunition.	Hull and machinery (1 + 2).	Ship complete (1 + 2 + 3 + 4).	Displacement, tons.	Cost per ton. Hull and mach.	Cost per ton. Complete.
	1	2	3	4	5	6	7	8	9
ENGLAND:									
B. S. *Superb*	2,354,600	1,215,900	2,123,000	2,277,400	3,570,500	7,970,900	18,900	189.1	420.7
B. S. *Vanguard*	2,161,600	1,061,500	2,123,000	2,451,100	3,123,100	7,797,200	19,600	166.0	397.6
B. S. *Hercules*	2,566,900	984,300	2,219,500	2,296,700	3,551,200	8,067,400	20,200	175.6	401.4
B. S. *Colossus*	2,566,900	1,061,500	2,219,500	2,277,400	3,628,400	8,125,300	20,200	179.5	403.4
B. S. *Thunderer*	3,184,500	1,042,200	2,412,500	2,528,300	4,226,700	9,167,500	22,900	185.3	401.4
B. S. *Conqueror*	3,319,600	1,080,800	2,412,500	2,335,300	4,400,400	9,148,200	22,900	193.0	399.5
B. S. *Monarch*	3,319,600	984,300	2,412,500	2,509,000	4,303,900	9,225,400	22,900	189.1	403.4
B. C. *Invincible*	2,663,400	2,103,700	1,833,500	1,852,800	4,767,100	8,453,400	17,500	272.1	482.5
B. C. *Inflexible*	2,702,000	2,026,500	1,833,500	1,717,700	4,728,500	8,279,700	17,500	270.2	472.8
B. C. *Indomitable*	2,740,600	2,065,100	1,833,500	1,814,200	4,805,700	8,453,400	17,500	274.0	482.5
GERMANY:									
B. S. *Westfalen*	2,991,500	965,000	2,026,500	1,524,700	3,956,500	7,507,700	18,500	214.2	405.3
B. S. *Rheinland*	2,991,500	965,000	2,026,500	1,524,700	3,956,500	7,507,700	18,500	214.2	405.3
B. S. *Posen*	2,991,500	1,061,500	2,026,500	1,524,700	3,956,500	7,507,700	18,500	214.2	405.3
B. S. *Helgoland*	3,512,600	1,158,000	2,412,500	2,373,900	4,670,600	9,457,000	21,650	216.2	436.2
B. S. *Thüringen*	3,512,600	1,158,000	2,412,500	2,373,900	4,670,600	9,457,000	21,650	216.2	436.2
B. C. *Von Der Tann*	2,895,000	2,123,000	1,833,500	1,158,000	5,018,000	8,009,500	19,000	264.4	422.7
B. C. *Moltke*	3,493,300	2,316,000	1,833,500	1,872,100	5,809,300	9,514,900	23,000	252.8	415.0

UNITED STATES:									
B. S. *Michigan*	2,895,000	965,000	1,408,900	1,930,000	3,960,000	7,198,900	16,300	235.5	442.0
B. S. *South Carolina*	2,933,600	965,000	1,408,900	1,930,000	3,960,000	7,237,500	16,300	239.6	443.9
B. S. *Delaware*	3,049,400	1,100,100	1,891,400	2,277,400	4,149,500	8,318,300	20,300	204.6	411.1
B. S. *North Dakota*	3,377,500	1,235,200	1,891,400	2,277,400	4,612,700	8,781,500	20,300	225.8	432.3
B. S. *Utah*	3,049,400	1,119,400	1,930,000	2,277,400	4,168,800	8,376,200	21,200	196.7	395.6
B. S. *Arkansas*	3,416,100	1,254,500	2,682,700	2,586,200	4,670,600	9,939,500	25,400	173.3	391.8
B. S. *Wyoming*	3,589,800	1,312,400	2,682,700	2,586,200	4,902,200	10,171,100	25,400	193.0	399.5
B. S. *Texas*	4,439,000	1,621,200	2,779,200	2,798,500	6,060,200	11,637,900	26,400	229.7	442.0
FRANCE:									
B. S. *Voltaire*	3,184,500	1,428,200	2,624,800	2,605,500	4,632,000	9,862,300	18,300	252.8	538.5
B. S. *Diderot*	3,165,200	1,428,200	2,624,800	2,605,500	4,612,700	9,843,000	18,300	250.9	536.5
B. S. *Condorcet*	3,145,900	1,428,200	2,624,800	2,605,500	4,593,400	9,823,700	18,300	250.9	536.5
B. S. *Vergniaud*	3,145,900	1,428,200	2,624,800	2,605,500	4,593,400	9,823,700	18,300	250.9	536.5
B. S. *France*	4,130,200	1,524,700	2,702,000	3,377,500	5,654,900	11,734,400	23,500	241.2	507.6
B. S. *Paris*	4,130,200	1,524,700	2,702,000	3,377,500	5,654,900	11,734,400	23,500	241.2	507.6
AUSTRIA-HUNGARY:									
B. S. *Viribus Unitis*	3,898,600	1,331,700	2,296,700	2,547,600	5,230,300	10,074,600	20,500	254.8	492.1
B. S. *Ersatz Erzherzog Friedrich*	3,898,600	1,331,700	2,296,700	2,547,600	5,230,300	10,074,600	20,500	254.8	492.1
B. S. *Ersatz Dandolo*	3,898,600	1,331,700	2,296,700	2,547,600	5,230,300	10,074,600	20,500	254.8	492.1
B. S. *Ersatz Saida*	3,898,600	1,331,700	2,296,700	2,547,600	5,230,300	10,074,600	20,500	254.8	492.1

ships have only four turrets and that the *Skoda* artillery is rather light, the cost of Austrian ordnance can not be considered moderate. Very high is the cost of machinery; and even the continuity of demand (which is, of course, a factor in the cost) has not been able to overcome the monopoly of the Stabilimento Tecnico Triestino.

With the exception of the United States, where the increased cost is due to the eight-hour clause, the general tendency is toward a slight decrease of cost. In France, particularly for the last two ships, there has been a sensible reduction. Armor is now much cheaper everywhere, and within the last two or three years this is true for turbine machinery. On the other hand, tin and copper cost more, and even steel is slightly higher, but this has very little importance in the total cost of a large warship.

In conclusion, a new ship of 25,000 tons complete, but without ammunition, could be obtained nowadays from shipbuilders of various nations at the following prices:

England	$10,100,000
Germany	10,400,000
United States	10,580,000
Austria	12,300,000
France	13,100,000

For the ship complete and ready for service we should add to these figures about $400,000 for general expenses (such as the Navy's designing staff, experimental tank, civil employés, inspectors, crews aboard ship during completion, etc.), and at least $2,000,000 for ammunition (in France, where a larger allowance is carried, an allowance of at least $2,500,000 should be made) and $200,000 for torpedoes, navigating instruments, etc., all of which, although taken from the Navy's stock, must nevertheless be replaced by a new supply when a new ship is ordered.

Therefore the taxpayers of various nations for each new ship of 25,000 tons need the following appropriations:

England	$12,600,000
Germany	12,900,000
United States	13,000,000
Austria	14,800,000
France	15,600,000

This shows at a glance how the Mediterranean nations are badly handicapped in comparison with other nations where the steel industry is more advanced and whose shipyards, owing to a larger and steadier output, are in better condition to give a return on the large capital invested.

NOTES.

THE LEAKAGE OF STEAM PAST PISTON VALVES.

By H. Denzil Lobley.

The important position which superheated steam is taking in modern engineering, and especially in its application to the semi-stationary type of engine, led the author to investigate the actual leakage of a piston valve as a possible source of loss, when using both saturated and superheated steam at various temperatures and pressures.

The results of these investigations are made all the more interesting when viewed in the light of Callendar and Nicolson's experiments of 1897, which pointed to the conclusion that the leakage of a slide valve was responsible for the greater part of the Missing Quantity.

The conditions on which the leakage of a slide valve depends were proved to be: Difference of pressure between the two sides of the valve, perimeter of the port, and the mean overlap of the valve.

Now, if the slide valve of an engine were replaced by a piston valve of the correct proportions to distribute the steam properly, these three conditions would remain unchanged, so that it seems reasonable to suppose that the laws of leakage which apply to a slide valve should also apply to a piston valve.

The following experiments were performed on an ordinary type of piston valve fitted with four Ramsbottom spring rings. The valve used was supplied by Richard Garrett & Sons, Limited, of Leiston, who have had many years of experience in the manufacture of piston valves. It was taken from stock, and was in no way specially treated, so that the experimental conditions would in no way differ from those when running in an actual engine.

The principle of the experiments is briefly as follows: The valve is driven up and down in a jacketed cylinder, at various speeds, by means of an electric motor. Steam is supplied from a boiler or superheater to the center of the valve, and the leakage is taken to a condenser, condensed into water, and weighed.

Fig. 2 is a section through the cylinder. The annular space between the liner and the cylinder forms a steam jacket which is connected to the steam-inlet pipe.

During the course of an experiment a steady flow of steam is blown through the jacket and out by the drain cock at the bottom of the cylinder. This prevents the stagnation and reduction of temperature, which would otherwise occur if the amount of steam used by the apparatus were only that of the leakage of the valve. It was found that by adjusting the flow through the steam jacket, the temperature in the cylinder could be kept very constant.

Protruding right into the steam space between the two portions of the valve is a thermo-couple thermometer, by means of which very accurate temperature readings can be taken.

A sight-feed lubricator is fitted, and this was adjusted so that in all experiments a constant feed of oil at the rate of one drop per minute was supplied. A cylinder drain cock is also fitted, so that when starting up all drainings from the apparatus and connecting pipes could be blown right through.

Fig. 1 shows the complete apparatus as used in experimental work. It

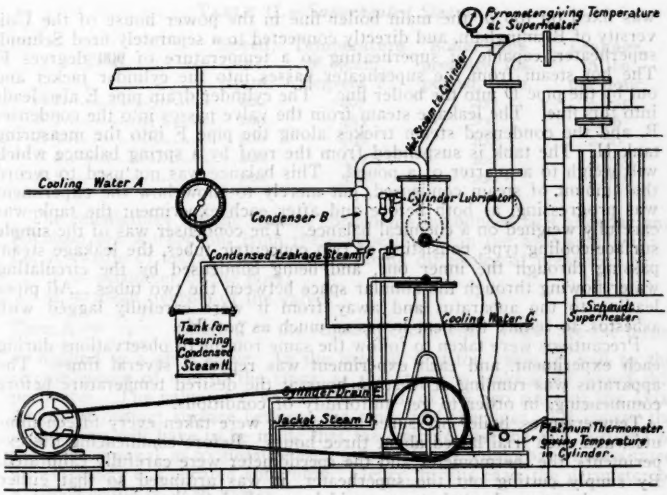


Fig. 1.—General Arrangement of Testing Plant.

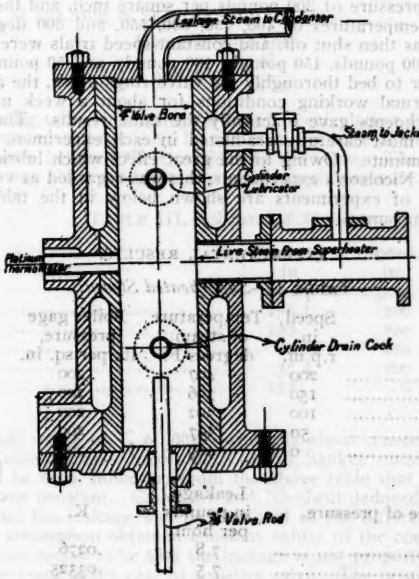


Fig. 2.—Section Through Cylinder.

was fixed on top of the main boiler flue in the power house of the University of Birmingham, and directly connected to a separately fired Schmidt superheater, capable of superheating to a temperature of 900 degrees F. The hot steam from the superheater passes into the cylinder jacket and out by the pipe D into the boiler flue. The cylinder drain pipe E also leads into this flue. The leakage steam from the valve passes into the condenser B, and the condensed steam trickes along the pipe F into the measuring tank H. The tank is suspended from the roof by a spring balance which will weigh to a quarter of a pound. This balance was not used to record the amount of steam condensed, but merely to show how the experiment was progressing, as both before and after each experiment the tank was carefully weighed on a chemical balance. The condenser was of the simple surface-cooling type, consisting of two concentric tubes, the leakage steam passing through the inner one, and being condensed by the circulating water flowing through the annular space between the two tubes. All pipes leading to the apparatus and away from it were carefully lagged with asbestos, to reduce the heat losses as much as possible.

Precautions were taken to follow the same routine of observations during each experiment, and each experiment was repeated several times. The apparatus was running at least an hour at the desired temperature before commencing, in order to get uniformity of conditions.

Temperatures, boiler pressures and speeds were taken every fifteen minutes, and each trial lasted about three hours. Before commencing the experiments, the thermometer and the speedometer were carefully calibrated. By simply cutting out the superheater, it was arranged so that either saturated or superheated steam could be supplied to the apparatus.

It was decided to commence by running experiments at various speeds with constant superheat and boiler pressure. The boiler was then kept at a constant pressure of 200 pounds per square inch, and the experiments were run at temperatures of 400, 450, 500, 550, and 600 degrees F. The superheater was then shut off, and constant-speed trials were run at boiler pressures of 200 pounds, 150 pounds, 100 pounds, and 50 pounds per square inch. In order to bed thoroughly the valve rings down, the apparatus was run under normal working conditions for about a week until two consecutive experiments gave practically the same results. The cylinder lubrication was most carefully graduated in each experiment to a feed of one drop per minute. Owing to the great effect which lubrication had on Callendar and Nicolson's experiments, this was regarded as very important.

The results of experiments are shown below in the tables with corresponding diagrams:

EXPERIMENTAL RESULTS.

TABLE I.—*Superheated Steam.*

Trial.	Speed, in r.p.m.	Temperature of steam, degrees F.	Boiler gage pressure, lb. per sq. in.	Leakage, in pounds. per hour.
A 1	200	407	200	4.825
B 2	150	396	200	4.82
B 3	100	392	200	4.9
B 4	50	397	200	4.8
B 5	0	409	200	2.4 (standing)

Difference of pressure.	Leakage, in pounds per hour.	K.	C.
141	7.8	.0276	.0011
113	7.5	.03325	.00132
65	6.6	.05075	.00202
22.4	4.9	.1095	.00436

TABLE II.—*Superheated Steam.*

Trial.	Speed, in r.p.m.	Temperature of steam, degrees F.	Boiler gage pressure, lbs. per sq. inch.	Leakage, in lbs. per hour.
A 1	200	407	200	4.825
A 2	200	452	200	3.66
A 3	200	498	200	2.70
A 4	200	557	200	3.075
A 5	200	588	200	3.275

The calculations for the leakage coefficients introduced by Callendar and Nicolson are worked out from these experiments in the following way:
Let K = rate of leakage per hour per pound difference of pressure;
P = perimeter of the port;
L = mean overlap of the valve.
Then $K = \frac{CP}{L}$ where C = the required coefficient. These are the terms as applied to a slide valve. In the case of a piston valve, P will be the circumference of one of the rings, as the steam is leaking all round this perimeter. The value of L will be the width of the two rings or the axial length of contact surface between the rings and the liner. Of course the assumption will have to be made that the leakage past the two portions of the valve is equal, as to measure each independently would have necessitated an extremely complicated apparatus. Hence, in this case, the perimeter of a 4-inch valve is 12.56 inches.

$$\text{or } P = 12.56 \text{ inches.}$$

The width of one ring is ¼ inch.

$$\text{Hence } L = .5 \text{ inch.}$$

$$\text{and } C = \frac{K}{25.12}$$

The following table shows the values of C calculating for the variable-pressure saturated-steam trials; by the term pressure is meant the difference of pressure between the two sides of the valve. The condenser side of the valve was always subjected to atmospheric pressure.

TABLE III.—*Saturated Steam.*

Trial.	Pressure, in lb. per sq. in.	Speed, in r.p.m.	Leakage, in lbs. per hour.
C 1	156.1	200	7.8
C 2	128	200	7.5
C 3	80	200	6.6
C 4	37.4	200	4.9
C 5	152	0	3.86 (standing)

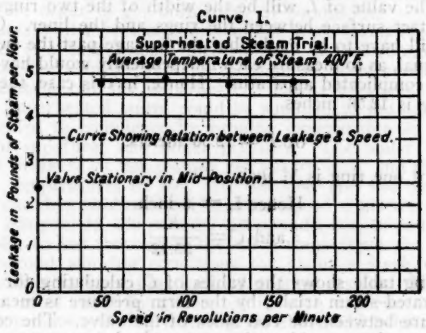
The average value of C is .0022, which is about one-tenth of that obtained by Callendar and Nicolson. Captain Sankey obtained a value of .003. It can be seen, however, from the above table that the value of C is by no means constant. Callendar and Nicolson deduced from their experiments that the leakage was proportional to the difference of pressure, and on this assumption obtained constant values of the coefficient C. The only conclusion seems to be that the leakage is not proportional to the difference of pressure in the case of a piston valve. The results of the experiments are very clearly shown in three plotted curves, speed leakage, pres-

sure leakage, and temperature leakage. From these results some very interesting conclusions can be drawn.

The great importance of the question of valve leakage is, no doubt, due to the probability that it may be responsible for a considerable portion of the missing quantity in a steam engine. In fact, Callendar and Nicolson's experiments pointed to this conclusion. The author's experiments, at any rate in the case of a piston valve, show this to be impossible. A 4-inch piston valve would in a semi-stationary type of engine deal with 1,500 pounds of steam per hour. Taking good average figures the missing quantity would be about 25 per cent. of this, or 375 pounds per hour. The valve when running at a temperature of 557 degrees F., only leaks 3.075 pounds of steam per hour; hence it seems quite safe to conclude that we must look somewhere else than in valve leakage for the missing quantity.

The effects of temperature on leakage show some very remarkable results. By superheating the steam only just sufficiently to dry it, the leakage is decreased from 7.8 pounds to 4.8 pounds per hour. This is an enormous reduction, and is a very important factor in support of the theory of condensation and re-evaporation. Assuming this theory, let us follow the action of the steam on the valve, and see how superheating would decrease the leakage. The space between the two portions of the valve being con-

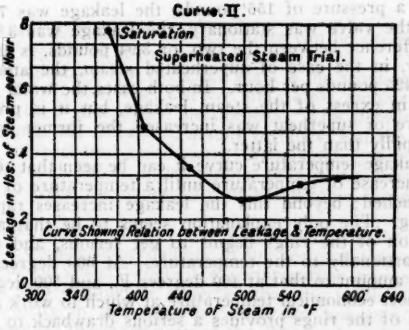
Curve. 1.



stantly in contact with hot steam is kept at a high temperature, whilst the exhaust sides being open to atmospheric pressure are comparatively cold. Consequently, as soon as the valve begins to move, it uncovers a cold portion of the liner upon which hot steam is immediately condensed. This condensation is repeated during the whole stroke of the valve, which on returning finds the liner coated with minute particles of water that are kept in this condition during half a revolution by the surface tension of the oil film. These particles of water then allow the rings to slide over them, and on emerging on the exhaust side of the valve, evaporate again into steam. It can easily be seen that if this is what happens, then the leakage will be decreased by superheating, as the hot steam gives up its superheat to the walls before it condenses, and thus condensation is avoided. If valve leakage really takes place in this manner, it should be affected by the speed of running, *i.e.*, the leakage should decrease with increase of speed, as then less time is allowed for condensation and re-evaporation. Very probably a decrease in leakage would have been shown during the experiments if the speed had been run up to 800 or 1,000 revolutions per minute, as in the case of high-speed engines; but 200 revolutions per minute was the highest speed at which the valve could safely be run.

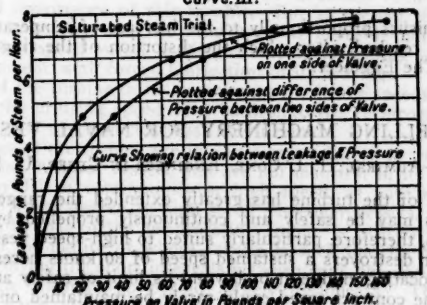
In order to facilitate this investigation it would be better at this point to divide the leakage into two distinct portions, (*a*) steam leakage, (*b*) water leakage. By steam leakage is meant that portion of the leakage

Curve. II.



which blows past the rings in the form of steam, through the minute fissures left between the rings and the liner owing to imperfect mechanical fitting; and by water leakage is meant that portion of the leakage which takes place in the form of water according to the theory of condensation and re-evaporation. That these two forms of leakage exist is shown by the experiments which were performed when the valve was stationary in mid-position. With saturated steam at a pressure on the valve of 152 pounds per square inch, the leakage with the valve stationary in mid-position is 3.86 pounds per hour. With the valve still stationary in mid-position, and a superheated temperature of 409 degrees F., only just sufficient to dry the steam, the leakage is decreased to 2.4 pounds per hour. These are cases of steam leakage only, as, under the circumstances, no other form of leakage is possible. If it were possible to so construct and fit a valve that the rings were in contact at every point with the liner, then this form of leakage would be eliminated. At first sight it seems peculiar that there should be any difference between the two above cases, as the area of the fissure between the rings and the liner should be unchanged, and hence the quantity of steam blowing through should be independent of the temperature. The area of the fissure remains the same, but the reduction of the leakage by the application of superheated steam is no doubt due to the fact that the latter has a less density than saturated steam, and hence a less weight of steam would be blowing through. As soon as the valve begins to move then water leakage takes place in addition. For instance, in the case of saturated steam, when the valve is running at 200 revolutions

Curve. III.


Saturated Steam Trial.
Plotted against Pressure on one side of Valve.
Plotted against difference of Pressure between two sides of Valve.
Curve Showing relation between Leakage & Pressure.
Leakage in Pounds of Steam per Hour.
0
2
4
6
8
0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150 160
Pressure on Valve in Pounds per Square Inch.

per minute at a pressure of 156 pounds, the leakage was 7.8 pounds per hour. When the valve was stationary, the leakage was 3.86 pounds per hour. The difference between the two, or 3.94 pounds, is the water leakage. Similarly, in the case of superheated steam, the amount of water leakage was 2.425 pounds per hour. In both cases the water leakage seems to be slightly in excess of the steam leakage, but it is probable that as the temperature of superheat was increased the former would decrease much more rapidly than the latter.

From the leakage-temperature curve it can be seen that the leakage decreases with increase of temperature until a temperature of about 500 degrees F. is reached; beyond this the leakage increases rapidly. This is very interesting. The only explanation seems to be that at 500 degrees F. the distortion of the rings begins to get serious, and this distortion increases proportionally to the temperature. At 600 degrees F. the leakage is equal in amount to that at 460 degrees F., and 500 degrees F. seems certainly the most economical temperature at which to work a piston valve. This distortion of the rings provides a serious drawback to the utilization of very high-temperature steam, as at high temperatures, when we should expect to get more economical water consumptions, we should probably get very serious valve leakage.

As stated at the beginning of this article, the leakage of a slide valve should be the same as a piston valve. A piston valve is simply a slide valve bent into a circle, and its advantage lies in the fact that being a balanced valve, less power is required to drive it, thus enabling it to be used in connection with automatic cut-off governors. At the time of the adoption of piston valves, the greater advantage which accrued from the reduction of valve leakage was not recognized. The great difference between the leakage of slide and piston valves is probably due to the fact that the former warp, and thus lift off the face. This is almost proved by Callendar and Nicolson's experiments, in which they show that the leakage of a slide valve is proportional to the pressure on the valve. Now, the increase of pressure of saturated steam is proportional to increase of temperature, which would cause a proportional increase of warping, and hence a proportional increase of leakage. Warping is eliminated except at high temperatures in a piston valve, owing to its shape, and hence the leakage is very small.

The principal conclusions of the author's experiments seem to be as follows: (*a*) Piston-valve leakage is not responsible for any appreciable amount of the missing quantity, or the leakage of a well-fitted piston valve is practically negligible; in fact, it is so small that one wonders where the saving of the various forms of patent piston rings on the market can possibly be. (*b*) The leakage does not follow the law $K = \frac{CP}{L}$. (*c*) The leakage diminishes proportionally to the increase of temperature until 500 degrees F. is reached, after which the distortion of the rings causes it to increase.—"The Engineer."

PROPELLING MACHINERY FOR NAVAL VESSELS.

By Rear-Admiral H. I. Cone, Engineer-in-Chief, U. S. Navy.

The advent of the turbine has greatly extended the range of speed at which vessels may be safely and continuously propelled by steam machinery. It is, therefore, particularly suited to high-speed vessels.

In the older destroyers a sustained speed of 30 knots necessitated driving the reciprocating engine at or beyond its limit of safety and reliability. As a result the contract speed could rarely be maintained on service, and certainly not for any extended period of time.

Today the steam turbines of our destroyers can be driven continuously at full power as readily as at the lower powers.

Next below the destroyer in the speed scale is the scout cruiser; of these we have three, the *Birmingham, Chester* and *Salem,* similar in design, except as regards their machinery installations. The *Birmingham* has reciprocating engines, the *Chester,* Parsons turbines, and the *Salem,* Curtis turbines. The contract-trial speeds were: *Birmingham,* 24.325 knots; *Chester,* 26.52 knots; and *Salem,* 25.947 knots, thus demonstrating, as in the case of the destroyers, the superiority of the turbine for development of high speed.

A rather unexpected result of our experience with these vessels on service has been the greater reliability of the reciprocating engine. In the case of this type of vessel, where the reciprocating engine is admittedly out of place, it was confidently expected that the turbine would demonstrate its superiority in this regard. Thus far, however, under similar conditions of service, the machinery of the *Birmingham* has required considerably less repairs than that of either turbine vessel, and of the three, the *Birmingham* has always been the one whose availability for service could be most be relied upon.

Thus for the scout cruiser, while the turbine is the most suitable type of machinery, its superiority over the reciprocating engine is not so manifest as in the case of the faster destroyer.

As we descend the speed scale into the region of the more important naval vessels of medium speed, the turbine is still less clearly indicated as the best type of propelling engine. This is because of the relatively increased importance of its principal deficiencies which are:

(1) A tendency to disablement to a degree requiring repairs beyond the capacity of the ship's force, thus temporarily destroying the military value of the vessel.

(2) A lack of economy of steam consumption at the lower speeds, which affects the cruising radius of the vessel.

Regarding the first of these, there is a tendency to a rubbing and stripping of the turbine blading, due to the small clearances necessarily employed, the causes of which are sometimes obscure, and the prevention of which is accordingly difficult.

The reblading of a damaged turbine can be done only at a navy yard, and requires the lifting of the casing and the rotor, which in itself is a considerable item. In our experience this work has seldom been completed within a month.

Improvement in design, greater experience with the operation of the turbine, and better facilities for its repair, will doubtless reduce somewhat the extent and importance of this trouble. But at present the reliability of the turbine in service must be considered as probably not as good in a medium or low-speed vessel as that of the reciprocating engine in its present development, with forced lubrication of its bearings.

Inferiority of the turbine in its economy at the lower speeds is probably inherent. At the cruising speed of a battleship, the reciprocating engine is 20 to 30 per cent. more efficient than the turbine.

These failings of the turbine are due in a great degree to the conditions governing its use in naval vessels, and are not manifest in its other applications. When driven continuously at high speed, as with electric generators, and, to a less degree, in high-speed passenger vessels, its efficiency as compared with that of the reciprocating engine is entirely satisfactory, and in a vessel the nature of whose service is such that the turbines can be kept warm during the short intervals between runs, the danger of stripping during the warming-up period largely disappears. The irregular movements of naval vessels, however, with their frequent periods in port with cold turbines and their considerable cruising at low speeds, impose handicaps on the turbine in these two respects.

The peripheral speed of turbines for maximum efficiency should be not less than half the velocity of the steam. To secure this maximum efficiency within practicable limits of diameter of the turbine requires, ordinarily, revolutions per minute of from 1,500 to 3,000. These speeds are perfectly practicable for electric generators, but are unsuited to the marine propeller, whose efficient working requires revolutions, depending upon its diameter, of from 70 to 800 per minute.

There is presented, therefore, the problem of providing a method in which a high-speed turbine may be made to drive a slowly-revolving propeller, thus conserving both turbine and propeller efficiencies.

Two methods of accomplishing this are particularly prominent just now.

Reduction-Gear Drive.—The collier *Neptune*, built by the Maryland Steel Company at Sparrow's Point, Maryland, has been equipped with reduction-gear machinery constructed by the Westinghouse Machine Company. This collier is a twin-screw vessel, displacing 19,300 tons, with a speed of 14 knots. Steam at a pressure of 200 pounds is supplied to a Westinghouse-Parsons turbine for each shaft, each turbine developing about 3,600 shaft horsepower on 1,220 revolutions. Between the turbine and its propeller shaft is mechanical gearing which will reduce the propeller speed to 135 revolutions per minute at about 14 knots. The gear pinions are carried in floating frames supported by oil pistons in such manner as to effectually take up any irregularity in the gear cutting. The oil pressure on pistons is indicated on gages which afford a means of directly reading the tooth pressure, and thus calculating the power being developed. A novel feature of the installation is a provision for operating the propelling machinery from the bridge. The shop tests of the gearing for the *Neptune* have been very satisfactory. The gears have demonstrated an efficiency of transmission in excess of 98 per cent., and have run quietly for long periods with no apparent wear. There is no back lash on reversing.

Extensive trials of the *Neptune* will be held.

Reduction gearing as installed in the cargo steamer *Vespasian* by the Parsons Company has successfully completed a year's trial, during which its durability has been tested under the severe conditions of weather in the North Sea during the winter months. The vessel has made thirty-two trips, a total of about 20,000 miles, carrying coal from the Tyne to the Continent and returning with water ballast. It is reported that during this time there has been no appreciable wear in the teeth of the gearing, and that there is practically no slackness apparent, although no provision was made of adjustment for wear. A noticeable feature of the installation has been the almost entire suppression of the tendency to racing of the engine, even in weather when the propeller has been entirely out of water.

Electric Propulsion in a Naval Collier.—In the naval collier *Jupiter*, a vessel similar to the *Neptune*, which is being constructed at the navy yard, Mare Island, California, there will be obtained a trial of a method of electric propulsion. A contract for this machinery has been placed with the General Electric Company. In this system it is proposed to install one turbo-generator running at a maximum speed of about 2,000 revolutions per minute, and delivering current with a potential of 2,300 volts to induction motors, one on each propeller shaft. The power from the turbine will be transmitted to the propeller shafts, electrically, with a speed reduction in ratio of 18 to 1. The efficiency of transmission is expected to be about 91 per cent. The water rate for this installation is stated to range from 12.15 pounds per shaft horsepower at 14 knots, to 15.55 pounds at 10 knots.

In the case of a battleship the proposed form of electric drive contemplates two turbo-generators driving two motors on each of the two main shafts. This multiplicity of units combined with a pole-changing

device on two of the motors, promises an economy close to the maximum at all speeds above 10 knots.

Comparing the electric with the mechanical reduction gear it is at once evident that the latter has the superior efficiency of transmission. In the case, however, of a vessel in which economy at the lower speeds is required, the flexibility of the electric installation, as outlined for a battleship, might overcome this disadvantage.

As with the scout cruisers, we will, therefore, probably have in service three similar colliers with different types of propelling machinery, the *Cyclops* being already in commission with an excellent installation of reciprocating engines, which on trial developed a horsepower on less than 1.5 pounds of coal per hour at full power.

Combination of Turbines with Reciprocating Engines.—The combination of a reciprocating engine working the steam from boiler pressure down to nearly atmospheric pressure, and a turbine which works from the exhaust pressure of the reciprocating engine down to the vacuum, is a particularly efficient one, because under such conditions each engine is working in the pressure range to which it is best suited. Such combinations are used with great success in power plants on shore and in certain moderate-speed passenger vessels, notably the *Olympic, Titanic, Laurentic* and *Otaki*. Its principal value is in economy of steam consumption at full power in a medium-speed vessel, in which case it probably gives a gain in economy over the reciprocating engine or turbine of not less than 10 per cent. In the case of naval vessels whose cruising speed requires only from 1/7 to 1/12 of full power, the advantage of this combination is not so impressive.

There is, however, a combination of reciprocating engine and turbine which has its maximum efficiency at the cruising speed. This consists in the use of a small reciprocating engine in a manner similar to that of the cruising turbine in a Parsons turbine installation. The reciprocating engine, whose size is limited to that required to pass the steam for the designed cruising speed, exhausts through the turbines. At higher speeds the reciprocating engine is disconnected. The weight of the reciprocating engine, and the space required, are less than for the cruising turbines, and it is probable that the engine will be less prone to disablement than the cruising turbine. The principal difficulty involved seems to be the design of the clutch connecting the reciprocating engine with the turbine shaft.

In another combination which suggests itself, and which, theoretically, has about the same efficiency, there is substituted for the reciprocating cruising element a small high-speed turbine connected by reduction gearing to the main turbine shaft.

The selection of machinery for a naval vessel at the present time, therefore, requires a careful consideration of a number of possible types.

Heavy-Oil Engine.—It now seems probable that none of the foregoing methods of propelling medium-speed naval vessels, all of which are forms of steam machinery, will endure. This in consequence of the recent remarkable development of heavy-oil engines of the Diesel type in Europe. Hitherto oil engines have not merited much consideration for large naval vessels on account of the limited power that could be developed in a single cylinder. An installation of any considerable power required a multiplicity of cylinders. Now, however, we are credibly informed that 1,000 horsepower has been developed in a cylinder about 33 inches in diameter with a 40-inch stroke, at 150 revolutions per minute, in a 2-cycle marine-type readily-reversible engine. This engine has a speed control that is satisfactory, and an economy of fuel consumption probably twice that of a steam engine. An experimental double-acting marine engine, with three cylinders, of about this same size, has been built to develop 6,000 brake horsepower at 150 revolutions per minute. In the development of the oil engine the trunk piston is being abandoned and crossheads and connecting rods used. It will, therefore, resemble the reciprocating steam

engine, and our experience with the latter will prepare us for the operating and repairing of the oil engine.

As in the case of all first installations, of whatever type, there will probably be disappointments in the first applications of the oil engine in marine propulsion, and it may be that further development of the engine may be found necessary prior to its extensive adoption. However, the superior economy of the oil engine, together with the elimination of the steam boiler and condenser will cause us to be patient in the face of a considerable amount of trouble.

The extent of the adoption of the oil engine by the nations of the world will be largely affected by the supply of oil available. We are in a particularly fortunate position in this respect, producing, as we do, two-thirds of the world's supply.

Use of Fuel Oil in Place of Coal.—In our new construction we are greatly extending the use of oil in steam vessels. All our new destroyers are oil burners, and it is hoped to extend its use to the larger vessels, including battleships. Some of the advantages of oil as compared with coal are as follows:

With oil an evaporation per pound of fuel greater than with coal in the ratio of about 14 to 9, and per square foot of heating surface in about the ratio of 10 to 8, is obtained. Fuel oil can be taken aboard readily without manual labor and without interruption to the routine of the ship.

Steam for full power in such a vessel can be maintained as readily as for low power. A vessel burning oil is capable of prolonged runs at full speed limited in length only by the supply of fuel. There is no reduction in speed due to dirty fires or to difficulty in trimming coal from remote bunkers.

There are no cinders and the amount of smoke can be controlled.

A considerable reduction in personnel is possible, the size of the fireroom force required being limited to that necessary for overhaul and cleaning of the plant.

The weight and space required for boilers is reduced, first, by the reduction in heating surface required, and second, by the shortening of firerooms, no space being required in front of boilers for handling fire tools.

Coal and ash-handling gear is eliminated. This renders unnecessary the piercing of the hull for coal trunks and for discharges from the ash expellers or ash ejectors.

The stowage and handling of oil is much easier than of coal, and will result in a much cleaner ship and consequent increase in time available for drills.

The mechanical supply of fuel to the boilers gives a prompt and delicate control of the steam supply, permitting more sudden changes in speed than with coal, which is a tactical advantage.—"Scientific American."

PROPERTIES OF AIR AND STEAM MIXTURES IN RELATION TO CONDENSING PLANT.

By THOMAS B. MORLEY, B.Sc., A.M.I. Mech. E.

Of recent years much attention has been given to the subject of the effect, on the performance of condensing plant, produced by air mixed with the exhaust steam. It has been recognized that even if leakage of air into the exhaust pipe and condenser were absolutely prevented, there would still pass into the condenser air which had entered the boiler dissolved in the feed water, and had passed through the engine with the steam, so that in all cases it is not merely steam that the condensing plant has to deal with, but a mixture of steam and air.

The effect of air on the behavior of the condenser and air pump (or pumps) is very marked, especially if the leakage of air into the exhaust is large, as it is in a carelessly jointed pipe line; hence it is important to have a clear knowledge of the properties of air and steam mixtures at the temperatures and pressures usual in condensers.

Several papers dealing with experiments on condensers and condenser problems have been published, and some have included diagrams embodying data for air and steam mixtures. Usually in these papers, however, the explanation of the principles and of the method of constructing the diagrams has not been very full, and the object of this article is to explain the method of calculating the data used in constructing one of the most useful of such diagrams, and to show its application.

Although no claim for absolute originality is made for the accompanying diagram, it may be remarked that it combines information not (so far as the writer is aware) previously collected into one diagram, and that the data have all been recalculated, the properties of steam being taken from Marks and Davis' "Steam Tables."

The fundamental principle (attributed to Dalton, and sometimes called "Dalton's law") regarding mixtures of gases or vapors is that the pressure of the mixture is the sum of the pressures of each of the constituents, each of which fully occupies the space containing the mixture. For example, 1 cubic foot of air and steam, consisting of 1 cubic foot of steam at, say, 90 degrees F. and 0.696 pound per square inch absolute pressure, and 1 cubic foot of air at 90 degrees F. and 0.778 pound per square inch will have a total pressure of $0.696 + 0.778 = 1.474$ pounds per square inch, which is equivalent to 3 inches of mercury, or the mixture is at a pressure corresponding to a vacuum of 27 inches with barometer at 30 inches. The pressures of the constituents are called "partial pressures."

The temperature is uniform throughout (equilibrium being assumed) and the "partial pressure" of the steam is that corresponding to the temperature, and is independent of the presence of the air.

Unlike a pure saturated vapor, a mixture of steam and air may change in temperature without changing in pressure, the temperature depending on the relative proportions of steam and air, on which the "partial pressure" of the steam depends.

The relation between the temperature, the pressure and the ratio of air to steam by weight may be found as follows:

Consider a quantity of steam and air containing 1 pound of steam, and let the temperature of the mixture be t degrees F., and the pressure p pounds per square inch absolute. Let the saturation pressure of steam at t degrees F. be q pounds per square inch absolute, and let the specific volume of the saturated steam be V cubic foot per pound.

Then the mixture consists of 1 pound of steam at a partial pressure q pounds per square inch, temperature t degrees F., and occupying V cubic foot, together with a quantity of air at a partial pressure $(p-q)$ pounds per square inch, temperature t degrees F., and also occupying V cubic foot.

Now at 60 degrees F. and 14.7 pounds per square inch, 1 pound of air occupies 13.1 cubic feet, therefore 1 pound of air at t degrees F. and $(p-q)$ pound per square inch occupies

$$\frac{14.7}{p-q} \times \frac{t+460}{60+460} \times 13.1 = 0.37\,\frac{t+460}{p-q} \text{ cubic feet.}$$

Hence the V cubic foot of air in the mixture under consideration weighs $\frac{(p-q)\,V}{0.37\,(t+460)}$ pounds, and the ratio R, say, of air to steam by weight is given by the equation

$$R = \frac{1}{0.37}\,\frac{(p-q)V}{t+460} = 2.7\,\frac{(p-q)\,V}{t+460}.$$

If pressure be expressed in inches of mercury, the formula becomes

$$R = 1.325 \frac{(p-q)V}{t+460}.$$

Example.—Let the mixture be at 27 inches vacuum (with barometer at 30 inches), and temperature 90 degrees F.

From steam tables the pressure q, corresponding to 90 degrees F., is 0.969 pound per square inch, or 1.417 inches of mercury, and V is 469.3 cubic feet per pound. The pressure p of the mixture is 3 inches of mercury; therefore

$$R = 1.325 \frac{(3-1.417)\ 469.3}{90+460} = 1.75.$$

Since the expressions giving q and V in terms of t are very complicated, it is difficult to work the above equation to find the condition of a mixture if R and p are the given data.

The most satisfactory plan is, for particular values of t, to connect values of R and of p with some other quantity having a simple relation to both, and to collect the results in the form of a diagram. Such a connecting link between R and p is provided by the volume of 1 pound of saturated air—u, say. Using notation as before, 1 pound of air has a partial pressure $(p-q)$, and, as shown in the previous calculation, its volume is

$$0.37 \frac{t+460}{p-q} \text{ cubic feet.}$$

(p and q being in pounds per square inch)—*i. e.*,

$$u = 0.37 \times \frac{t+460}{p-q} \text{ cubic feet.}$$

If the pressure is expressed in inches of mercury, the formula becomes

$$u = 0.755 \times \frac{t+460}{p-q} \text{ cubic feet.}$$

It must be noted that u is the volume of 1 pound of air, not of 1 pound of mixture—*i. e.*, it is the air alone in the volume u that weighs 1 pound.

Example.—Let t be 90 degrees F.; p, 27 inches vacuum, or 3 inches of mercury.

From steam tables $q = 0.696$ pound per square inch, or 1.417 inches of mercury; therefore

$$u = 0.755 \times \frac{90+460}{3-1.417} = 262 \text{ cubic feet.}$$

Values of u are calculated for a series of values of p and t, and the diagram is constructed by choosing temperatures as abscissæ and plotting values of u as ordinates; the full-line curves correspond to particular values of the pressure.

The diagram should, of course, be drawn upon squared paper.

It now remains to connect values of the ratio R with values of u. We have already found

$$R = \frac{1}{0.37} \frac{(p-q)}{t+460} V,$$

and

$$u = 0.37 \frac{t+460}{p-q};$$

therefore

$$a = \frac{V}{R}.$$

Example.—Let the temperature be 90 degrees F., and ratio R of air to steam be 2. V corresponding to 90 degrees F. is 469.3 cubic feet per pound; therefore

$$a = \frac{469.3}{3} = 234.7 \text{ cubic feet.}$$

Plotting the results of such calculations, the dotted lines on the diagram are obtained for particular values of the ratio R. The diagram not only represents the properties of air and steam mixtures, but is capable of useful applications in the solution of condensing-plant problems.

The substance (other than the water of condensation in the case of a "wet"-air pump) passing from the condenser to the air pump is saturated air, and its volume is shown by the ordinates of the diagram. The

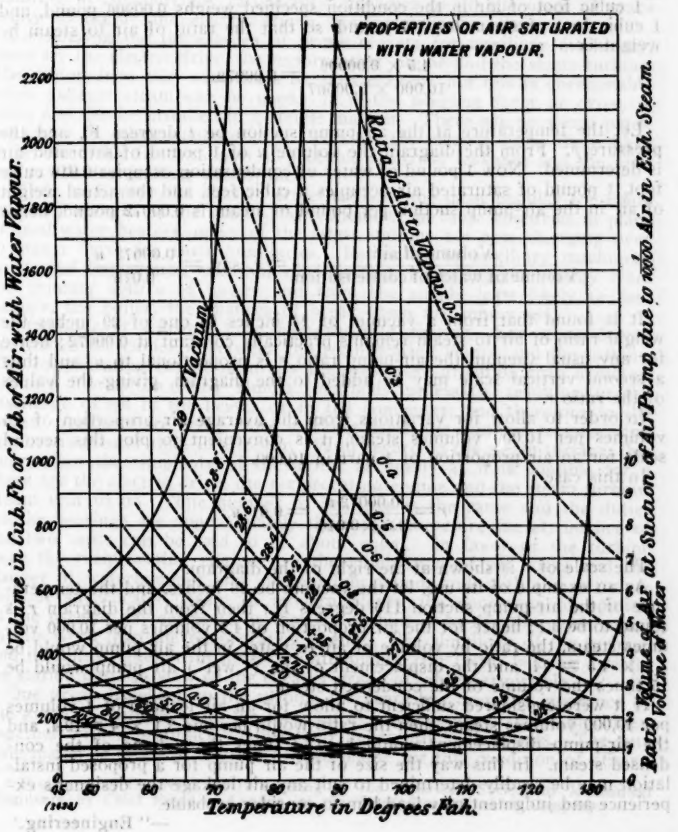


diagram illustrates (by tracing a constant-vacuum line) the great increase in air ratio and reduction in volume due to cooling the condenser discharge, and if a reasonable estimate of the amount of air likely to be present in the exhaust steam can be made, the diagram may be used to determine the size of the air pump.

Mr. T. C. McBride has shown that under average conditions and for 26 inches vacuum it is necessary to provide for 4.5 volumes of air per 10,000 volumes of steam in the exhaust pipe. To compare this with the diagram it is necessary first to find the weight ratio of steam and air. We have to compare the weight of 4.5 volumes air at 26 inches vacuum and 125 degrees F. with that of 10,000 volumes steam at the same temperature and pressure. [125 degrees F. is the temperature of saturated steam at 26 inches vacuum. The presence of air in usual quantities is insufficient to lower appreciably the temperature of the mixture below saturation temperature until condensation is almost complete, so that in the exhaust pipe the temperature is practically that of saturation corresponding to the vacuum.]

1 cubic foot of air in the condition specified weighs 0.00906 pound, and 1 cubic foot of steam 0.00567 pound, so that the ratio of air to steam by weight is

$$\frac{4.5 \times 0.00906}{10{,}000 \times 0.00567} = 0.00072.$$

Let the temperature at the air-pump suction be t degrees F., and the pressure p. From the diagram the volume u of 1 pound of saturated air is determined. Now 1 pound of water of condensation occupies 0.016 cubic foot, 1 pound of saturated air occupies u cubic feet, and the actual weight of air in the air-pump suction per pound of steam is 0.00072 pound, hence

$$\frac{\text{Volume of air}}{\text{Volume of water of condensation}} = r, \text{ say,} = \frac{0.00072\ u}{0.016}.$$

It is found that from a vacuum of 24 inches to one of 29 inches the weight ratio of air to steam remains practically constant at 0.00072; hence for any usual vacuum the air-pump ratio r is proportional to u, and thus a second vertical scale may be added to the diagram, giving the values of the ratio r.

In order to allow for variations from the average air proportion of 4.5 volumes per 10,000 volumes steam, it is convenient to plot this second scale for an air proportion of 1 part in 10,000.

In this case

$$r = \frac{0.00072\ u}{4.5 \times 0.016} = 0.01\ u.$$

The scale of r is shown at the right of the diagram.

As an example of its use, let the vacuum be 26 inches, and the temperature of the air-pump suction 110 degrees F.; then from the diagram r is found to be 3.1; hence for the air proportion of 4.5 volumes per 10,000 volumes steam, the ratio by volume of air to water in the air pump would be $3.1 \times 4.5 = 14$, and the displacement of the ("wet") air pump would be 15 times the volume of the condensed steam.

If it were considered sufficient to allow for an air leakage of 4 volumes per 10,000 volumes steam, then the ratio would become $3.1 \times 4 = 12.4$, and the air-pump displacement would be 13.4 times the volume of the condensed steam. In this way the size of the air pump for a proposed installation may be readily determined to suit any air leakage the designer's experience and judgment may lead him to consider probable.

—"Engineering."

STATUS OF THE SMALL STEAM TURBINE.*

By W. J. A. London.

The new uses for the turbine and the fact that all builders of small turbines are turning out more work every year are sufficient evidence that there must be some very good reasons for this change in ideas. In studying the early history of the steam turbine, it will be found that the greatest factor tending to retard the advancement of the turbine was not the turbine itself, but the difficulty of finding a suitable and satisfactory machine that it could drive. The troubles experienced with generators in the early days are ancient history, and with the applications of the small turbine for auxiliary purposes, history is again repeating itself. The small turbine has been established for a long time, but many difficulties have been encountered in the satisfactory production of efficient generators, pumps and blowers in order to make the whole unit a commercial success. The makers of this apparatus have appreciated the indisputable advantages of the turbine drive and have laid themselves out to cater to this field, resulting in commercially satisfactory machines to fill the place of practically all engine and electric-driven auxiliaries in the past.

Considering the three types of prime movers for auxiliary machinery, there are the electric drive, the reciprocating engine and the steam turbine. The motor drive may sometimes be most efficient, but this is questionable when exhaust steam can be used. It is the simplest form of drive, is clean and can be installed in a simple manner. There are no steam pipes, pet cocks or other encumbrances. Against the electric drive there have to be faced the disadvantages of probable electrical leakages and the serious accidents resulting therefrom.

While in Europe the electric drive has been the most popular for many years on account of the use of the economizer in those countries in place of feed-water heaters, many of the latest stations are now changing over to steam drive for their auxiliaries. Reliability in auxiliary machinery is of vital importance, and it is obvious that, by taking the energy from the primary source of power and from the supply least likely to fail, namely, the boiler, the chances of break-down are minimized.

Advantage of Turbine over Reciprocating Engine.—For steam-driven auxiliaries the choice is between the reciprocating engine and the turbine. The reciprocating engine naturally requires more attention than a motor, cylinder lubrication being almost invariably necessary, resulting in the discharge of oil to the heater or to the condenser, and therefore to the boiler. Again, there being more working parts, the wear of reciprocating engines is greater than in a motor. With regard to the steam turbine, this approaches the simplicity of the motor as nearly as it is possible for a there are the electric drive, the reciprocating engine and the steam turbine. steam unit to do. The number of bearings is the same and the duties on the bearings are approximately the same; therefore the attendance of the two units can be said to be about equal. In favor of the turbine over the reciprocating engine, are the elimination of pet cocks and the danger of waterhammer and internal lubrication, and, with the introduction of superheated steam to modern plants, the consensus of opinion at the present day seems to be that the turbine is by far the more suited to this work than the reciprocating engine. The simplicity of the turbine, and its limited amount of necessary attendants, is a factor bearing directly on station costs in view of the cheaper grade of labor permissible.

One great drawback to the turbine for years has been the ignorance of the average engineer as to the mechanical construction of these machines, together with a prejudice against their apparently high speeds. The turbine maker is probably a great deal to blame for this condition of affairs,

* Presented at the New Haven meeting of the American Society of Mechanical Engineers, by Chief Engineer London, of the Terry Steam Turbine Company.

it having been the policy to keep the construction of the machines dark. A broader view, however, exists at the present time, and consequently engineers are becoming more experienced and now feel themselves competent to take care of these machines without having to send to the manufacturer for the smallest repair or adjustment that may be necessary.

Efficiency of Small Turbine.—With regard to the efficiency of the small turbine, both with regard to a comparison of different types and to a comparison with motors or reciprocating engines, the writer would like to enlarge on the real meaning of efficiency. The question of water rate does not cover the situation. In a 5,000-kilowatt turbine, the consumption of each pound of steam per kilowatt means probably $2,400 per year. In a small turbine, a difference of one pound per horsepower in a 100-horsepower machine means for the same period of operation, $48 per year.

Practically all small turbines on the market at the present time can be improved in water-rate efficiency. This can be done by increasing the speed in revolutions per minute or by increasing the peripheral velocity, using larger wheels, or by an increased number of stages. All these points mean additional complication, additional weight and higher first cost.

For the sake of argument, a single-stage turbine with a 2-foot wheel, if increased 3 inches in diameter will bring down the water rate from 38 to 36 pounds; in other words, a saving of 5 per cent. would mean an increase in weight of the whole turbine of about 17 per cent. The increase in floor space would be 10 per cent., and the increase in the cost or selling price would be approximately 15 per cent. The point then for the designer to consider is where to draw the line between absolute water rate and commercial efficiency.

It is obvious that a comparatively slow-speed machine will give less trouble, not only to itself but to the other end of the apparatus, and a breakdown due to over-sensitiveness on the part of the turbine of high efficiency will soon run away with the money saved by the increased efficiency.

For example, assume a 100-horsepower turbine of a slower speed to consume 2 pounds more per horsepower than another, which means an additional cost of about $96 per year. This $96 per year means 4 cents per hour, and the difference between the cost of unskilled labor necessary for the one machine and the skilled labor necessary for the attention of the other generally amounts to much more than this. Assuming the case of an exciter running a large engine, one hour's shutdown per year at the rate of 3 cents per kilowatt-hour will more than pay for this additional coal bill.

Supposing again the turbine is to run the condensing apparatus, the difference in steam consumption of a 5,000-kilowatt machine running over from condensing to noncondensing will take a running period of only six hours to pay for the $96 additional coal consumed. This question of reliability is, to the writer's mind, not sufficiently emphasized, and should be considered an absolutely direct part of the efficiency factor, and the average engineer is now sufficiently well versed in the construction of steam turbines to be able to determine for himself what he can expect in overall efficiency when given the water rates and the construction of the various machines.

Not a "Steam Hog."—With reference to the actual water rates obtained, the idea existing among many engineers that the turbine is a "steam hog" is entirely erroneous, as will be shown. Of course, examples can be found every day where conditions are such that if economy is a factor to be considered, poor judgment was shown in the first instance in ever installing a turbine. This state of affairs is not by any means confined to the small turbine; many examples of it may be seen in the larger marine installations.

In practice, the engineer does not care about turbine water rates. He

is interested in his overall water rates per air, electrical or water horsepower, as the case may be, and instead of always criticizing the turbine end, it is the other end that for many years has needed reforming.

Water-rate Efficiency.—The work of the builder of the other apparatus has resulted in a unit which can hold its own against all competition, not only in reliability but in actual water-rate efficiency. It has been a practice for years to run gas blowers at turbine speeds, but as there were no turbines in the earlier days, the belt drive was adopted. Later installations are using the direct turbine drive with the same blower speed as previously, which gives a better overall efficiency. A fair figure at the present time for a modern gas plant operating with 125 pounds steam noncondensing, with a turbine of 100 horsepower, is about 70 to 80 pounds per air horsepower with a blower pressure of 22 inches.

In some tests quoted by Mr. Morehouse in a paper read before the Southwestern Gas & Electric Co. on an engine-driven fan of 6,000 cubic feet capacity, with 20 inches pressure, the water rate per air horsepower was 141 pounds. Some recent tests on a turbine blower of slightly larger capacity, also operating under normal gas-works conditions, show an actual water rate of 76.2 pounds. In some of the later installations, higher blower pressure is being employed. Not only does the turbine efficiency in this case increase, but the fan efficiency also. In a recent test carried out on a 40-inch blower for the Brooklyn Union Gas Company, 58.5 pounds per air horsepower was obtained, with a steam pressure at the throttle of 145 pounds noncondensing.

Further tests on this equipment show a turbine water rate, when operating with 117 pounds steam and 5 pounds back pressure, of 36 pounds per brake horsepower or a Rankine cycle efficiency of 49.8 per cent. This figure is remarkably good, but it only shows what can be done in other installations if the apparatus to which it is connected can be brought up to the same standard of efficiency as can the turbine at turbine speeds.

The question of generator efficiency is no longer an unknown quantity, and guarantees made on brake horsepower can readily be verified from the electrical readings. It is unfortunate that the same cannot be said with regard to pumps and blowers, and it is often found that where a guarantee is made by the turbine builder and it apparently falls down on an actual test with a blower or pump, the pump is not operating under conditions called for in the contract and it is therefore not running at anywhere near the efficiency specified.

Comparative Performances of Turbine and Engine-driven Pumps.—With regard to the comparative performances of turbine-driven and engine-driven pumps for boiler-feed purposes, some interesting comparisons are obtainable from the tests made on board the scout cruisers *Birmingham*, *Chester* and *Salem*. Together with these figures are given the water rates that are obtainable from the modern turbine boiler-feed pump, showing a marked increase in efficiency in favor of the turbine-driven sets. (See table.)

Water Rates of Small Turbines.

	Steam pressure, pounds.	Back pressure, pounds.	G. P. M.	Head, in feet.	W. H. P. Water rate.	Turbine, pounds.	Improvement, per cent.
Birmingham	183	6	329	488	83.0	61.8	25.5
Salem	203	6	193	690	91.8	61.2	33.6
Chester	187	6.7	219	610	101.0	64.0	34.6
					275.8	187.0	93.7
Average					91.9	62.4	31.2

The assumed pump efficiency equals 85 per cent. and the assumed engine efficiency equals 90 per cent. on reciprocating sets.

Napier's Formula.—On the general subject of the steam consumption of small steam turbines, it is strange that Napier's simple formula

$$W = \frac{PA}{70}$$

where

W=Weight of steam per second;
P=Absolute higher pressure;
A=Area of passage in square inches;

for the flow of steam through nozzles cannot be used to greater commercial advantage than it is at the present time. This well known formula has been proved so many times to be approximately correct that it is a pity that it is not more used to compute the steam consumption. The majority of small turbines now operate with a terminal pressure at the jet below 0.58 of the initial pressure, so that this formula can be readily applied to any machine and the steam consumption determined by placing a gage in the steam ring.

The maker could be called upon to give the number and the size of the jets, and in this way an instantaneous reading could be taken at any time showing the amount of steam passing at any one time. If this were done, less claims would be made that a turbine is passing a greater amount of steam than the makers claim for it. Of course, when it comes to a question of pump efficiency, this is a harder matter to determine, and it is possible that at a given water horsepower the turbine does require more steam than called for. This can only be tested out by means of a dynamometer or brake to find out where the trouble lies.

By a slight modification to the constant 70 in Napier's formula for various jets, figures can be obtained as accurate as is possible by actual measurement. In a series of tests carried out within the last few days on a certain machine, in seven readings with a ring pressure varying from 40 to 150 pounds, the minimum discrepancy between the calculated and the actual results was 1.3 per cent. and the mean over all the readings, 0.443 per cent.

The actual figures obtainable on test and those calculated are as follows:

Napier's Formula	Actual Tests	Per Cent. Variation
2,028	2,043.6	0.8
1,264	1,248.0	1.3
1,935	1,934.0	0.0
2,625	2,638.0	0.4
1,905	1,906.0	0.0.
3,047	3,066.0	0.6
2,272	2,272.0	0.0
		3.1

Average, 0.443

Simplicity of Design.—The construction and the general design of small turbines have been so thoroughly dealt with in other papers that it is not considered necessary to go into this phase of the subject. The writer would, however, like to bring forward the fact that in small turbines, as in the larger units, the designs and mechanical construction are all-important factors. What is considered good practice in the design of large turbines is, however, in many cases a mistake in small machines.

As to the details of the machine, the design should be as simple and the governor and other mechanism attached thereto as fool-proof as possible, even at the expense of efficiency if necessary. The wheels, etc., should be readily accessible, as facilities for handling such machines in the places where they are so often installed are very inadequate for this purpose, and the time allowed for the dismantling of an auxiliary is often only a few hours at the middle of the night.

That machines are designed to fulfill these requirements, and that complications in the design are unnecessary are proved by the constructions now employed. For example, it is thought by some engineers that the single governor mounted directly on the turbine shaft cannot be made sufficiently sensitive for close regulation. Figures taken from some recent government tests on Terry turbines show that this is by no means the case.

On a series of six consecutive 5-kilowatt machines for the Navy, the official figures were as follows: Maximum jump of the worst machine, with full load thrown off, 3.28 per cent.; average maximum jump over the six machines, 2.61 per cent., with a mean settled variation of 1.19 per cent.

On two 85-kilowatt sets, also for the Navy, the mean figure of the two machines were, maximum, 2.77; settled, 2.09.—"Power."

IMPROVEMENTS IN SUBMARINES.

By Henry Bernay.*

From the beginning of the *Dreadnought* era, which was characterized by a progressive and finally alarming increase in the displacement of ships of the line, we hear much less said about submarines. In a comparison of fleets they do not enter into the computation, and one would say that after having given them perhaps an exaggerated importance several years ago, an equally excessive reaction has caused them to be considered today, at least by some nations, as an auxiliary arm of only a limited interest.

With the increasing dimensions and speed of surface vessels, the automobile torpedo appears to have become less effective; because, on the one hand, the vessel's speed diminishes the chances of its being hit, and on the other, the increase in tonnage and more perfect watertight subdivisions tend to reduce the damage by explosion. But the automobile torpedo has also recently undergone great improvement. Its speed now exceeds 40 knots; that is to say, in the last twenty years it has increased proportionally with that of the fast ships. The accuracy of its trajectory has improved, and, finally, its charge has been considerably increased. However, we lack a demonstration of the effect of this charge on a modern hull, but, judging as we can from the test made at Lorient against a bulkhead of the *Mirabeau* its effect will be enormous, and there is no system of construction that could withstand it. While it is probably an exaggeration to anticipate the destruction of a modern battleship by a single torpedo, at least it might disable her for a time.

To the same extent as the weapon, so has the submarine, that has to fire it, been perfected. The opportunities that it will have for attack and the chances of success have become more numerous and more probable. These depend principally upon the speed of the submarine—surface and submerged.

A submarine can devote a much greater part of its displacement to its propelling machinery than most types of vessels. For though its hull, necessarily thick, weighs much, its armament is very light. On the first *Gustave-Zédé*, purely electric, the motors and storage batteries took nearly

* Translated from "Le Yacht," by B. Gregory, Associate.

50 per cent. of the displacement light. The more recent submarines have more weight of hull, having been built to withstand much greater pressures, but they still require for their propelling machinery between 35 and 45 per cent. of their displacements—as much as fast cruisers. This great weight is divided between two independent sets of motors, and therefore the speeds of submarines are not of the same order as those of ordinary vessels.

The submerged propelling machinery is the heavier. It takes, according to type, from 20 to 30 per cent. of the displacement. It is also the least efficient, because it is used when the hull is submerged, that is to say, when it offers the maximum resistance. Moreover, the number of storage-battery cells is limited, as it is impossible to increase their weight, which is already considerable, and, on the other hand, there are also questions of space and facility for overhauling, that must be considered. Nevertheless, the submerged speed has been much increased; that which was 6 to 7 knots has now reached 10 to 12 knots in the latest types. This increase is due to the constant and rapid improvement in storage batteries, which are now capable of discharging with safety at a rate one and one-half times greater than the old type. But it does not appear that we can gain much more. In order to do so a light storage battery of the Edison type would have to give results that up to the present time have only been obtained in laboratories.

High submerged speed, moreover, brings up other difficulties that make it appear impracticable for the present. In the first place the periscope standards, which vibrate excessively, must be made more substantial, thus rendering them more easily seen in themselves and by their wake. Furthermore, the submerged equilibrium, constant trim and immersion are more difficult to maintain and require horizontal rudders of larger dimensions. We are satisfied, however, that none of these difficulties is impossible to solve. The solution will be discovered when it is necessary, but we have not yet reached it.

It is this limitation of submerged speed, the tactical element essential to the use of the submarine, that is actually its weakest point. On the contrary the increase in surface speed is very great, and we can count on further improvement in the near future, because this progress is related to that of the oil engine, which is more marked each day.

Without going very far back we have a striking example by comparing the *Pluviose*, commenced in 1905, with the *Faraday*, that has just had its trials. The former, propelled by steam engines, did not exceed a maximum speed of 12.5 knots, while the latter, of the same tonnage but with oil engines, attained 15 knots. Vessels very nearly like the *Creusot*, built by foreign navies after plans by M. Laubeuf, should reach 16 knots with the same surface displacement of 400 tons and the submerged speeds should be slightly improved. Besides, the gain in cruising radius is very great and reaches almost to double that of the steam engine, thanks to the small fuel consumption of the oil engine.

We can expect from improvement in these engines an increase in surface speed for submarines, but it is well known that the requirements of those who use the engines are the principal stimulant to the designing engineers, which requirements are at this time particularly urgent. We want submarines capable not only to defend a short strip of coast like those that were first constructed, not only to make relatively long runs like the *Pluviose* and to operate at a great distance from their base, but at the same time to coöperate in fleet engagements on the high seas and to bring the full strength of their power to bear at the decisive moment, next to which the coast-defense consideration becomes secondary. But, to fulfil this rôle more speed and more endurance is necessary than has been obtained up to the present time.

It is this consideration that gave birth to the Navy Department's elab-

orate program in 1908 for seagoing submarines, imposing a trial surface speed of 20 knots, with the same radius of action as the torpedo-boat destroyers. It is known that this program gave rise to a number of propositions, of which that of the Engineer-in-Chief Simonot was adopted, and two-type vessels were placed under construction; the *Gustave-Zédé*, in 1910, and the *Néréide*, in 1911. They reached 740 tons surface displacement, which was doubtless a mistake, considering the cost, nearly double that of the *Faraday*. But, with submarines as in all classes of vessels, it is necessary to allow an increase in both tonnage and cost, according to requirements and qualities. The displacement of our torpedo-boat destroyers has varied during the same period and in the same proportion as that of our submarines, and we do not build more small vessels of one class than another. It is all a matter of comparison between the cost and the service to be performed by new vessels as compared to those that have preceded them.

The increase in displacement permits, not only to attain speeds heretofore impossible, but also to sustain these speeds under less favorable conditions, which is essential. The objections from a maneuvering standpoint will certainly be overcome without much difficulty. It was from this standpoint that troubles were feared by the jump from the *Pluviose*, of 400 tons, to the *Gustave-Zédé*, of 740 tons, as was likewise the case is passing from the 200-ton *Sirene* type to the 400-ton *Pluviose*, and which troubles experience has overcome. It suffices that designers take the necessary precautions to assure maneuvering qualities and security in spite of the increase in dimensions.

It is not only in France, however, that this idea prevails. In England a submarine of 800 tons surface displacement is under construction and the Admiralty has admitted this large displacement simply to attain a speed of 16 knots. This goes to demonstrate that the ideas on the requirements of submarines have evolved in nearly the same manner on both sides of the Channel, with the motor improvements more advanced with us. The Germans are also constructing a submarine of nearly equal tonnage. Be it noted, these English and German submarines will be provided with a gun of 75 mm. and 88 mm., respectively. It is, therefore, no longer admitted that a vessel of this importance could find itself without any means of defense and at the mercy of the smaller torpedo craft, if it were unable to submerge. This is an example that will certainly be followed.

But, if submarines must have greater speed and radius of action on the surface, more speed and radius of action submerged, more habitability and more guns, will they increase more in size? This will be an irresistible necessity, if we do not find a way to save weight in the propulsive machinery by using the surface motors when submerged.

We know that this has already been the object of investigation in the past, particularly in France. It was in 1902 that the submarine "Y" was placed under construction, which, when submerged, would operate its oil engines by means of compressed air released into the hull and then drawn into the engines. Special condensers were provided to accumulate the gases of combustion and to exhaust them at long intervals. The construction of this vessel was lengthy, its engines gave little satisfaction even when working under normal conditions, and when it was desired to run them with the vessel closed up, it was almost immediately necessary to give it up. The "Y" was condemned without seeing service. Another submarine, the *Omega*, was in the meantime constructed to operate on the same principle, only with a greater displacement. It is now steam propelled under the name of the *Argonaute*. At the same time we undertook the construction of two small submarines called, *Guêpe No. 1* and *Guêpe No. 2*, in which the oil engines operated in closed cycle, fed by oxygen. The engines were never delivered and the hulls have remained unfinished.

So many unfortunate experiments seem to have discouraged the French Navy. They served, however, to make an effort in a totally different direction, by undertaking in 1906 the construction of the *Charles-Bruin*, with Maurice steam-storage boilers. This submarine, of 350 tons, begun the 31st of December, 1906, has not yet made its trials and it is not known what it will do. Besides, it does not seem to furnish a complete solution of the single-engine submarine, because its speed endurance submerged is necessarily very limited.

But in foreign countries a complete solution of oil engines operating under water is being investigated. An Italian shipbuilding firm is actually constructing a submarine conceived on this principle, designed by M. del Proposto.

When this problem is solved it will result in the combining of the torpedo boat and the submarine, a combination that certain parts of the naval program of Admiral de Lapeyrère seem to anticipate for the near future. It appears premature, however, to expect it before improvements, as yet unheard of, are tried to overcome the too-apparent present difficulties. It is, therefore, necessary to content ourselves with perfecting the two types separately, by enlarging their offensive power and their possibilities in action.

EXPLOSION OF A SPANISH OMELET.

That Spain is still a little bit hostile to the United States was demonstrated to a girls' class in cookery in Washington Irving High School, at No. 142 West Twentieth street, yesterday, when the rude behavior of a Spanish omelet resulted in the building taking fire. Miss Emma Crane, who was giving the lesson, sounded the alarm and all the girls went out with the idea that they were going through a fire drill. It was not until they reached the street that they learned of the perfidy of the omelet.

The glorious American hen had provided her best offering for the omelet, the tomatoes had been introduced into the mixture, the Spanish onion was feeling perfectly at home, and so were the peppers. The blow-up came when the Irish potatoes were put in. They swelled with indignation, and of course the omelet swelled with them. It began to look to Miss Crane as if her pupils had compounded a felony instead of an omelet. Finally the mixture, led by the potatoes, and desiring liberty or death, burst from the sheathing of yellow and landed on walls and ceiling. Much of the material landed against a blackboard on which the prescription for the meal was written.

The damage to the schoolroom itself is about fifteen cents, fully covered by insurance.—"The Locomotive."

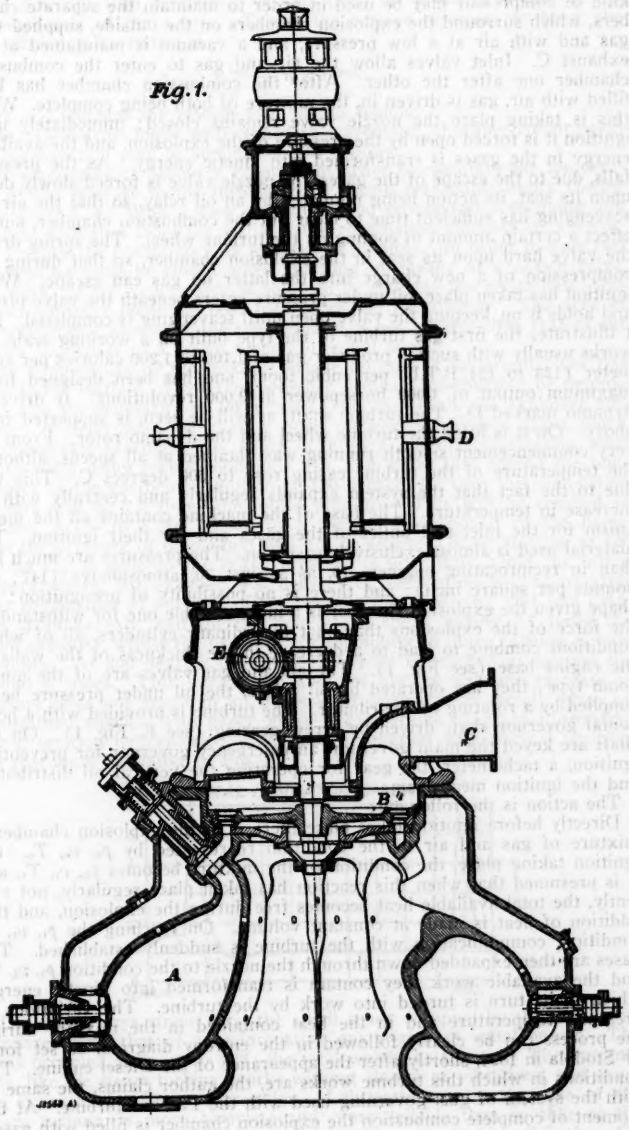
THE GAS-TURBINE.*

By H. Holzwarth, Mannheim.

The turbine described is an explosion turbine. The combustion chamber, marked A in Fig. 1, is intermittently filled with the gas and air mixture; then follow ignition, explosion, and increase in pressure of the burnt gases and their expansion through the nozzle, after which they act on the turbine wheel marked B. The explosion chamber is next effectively scavenged and cooled by fresh air. A number of similar explosion chambers are arranged in a circle at the base of the turbine, and act one after the other. Any

* Abbreviated translation of a paper read in German before the meeting of the Schiffbautechnische Gessellschaft, held in Berlin, November 23 to 25.

Fig. 1.



kind of compressor may be used in order to maintain the separate chambers, which surround the explosion chambers on the outside, supplied with gas and with air at a low pressure, and a vacuum is maintained at the exhaust C. Inlet valves allow the air and gas to enter the combustion chamber one after the other. After the combustion chamber has been filled with air, gas is driven in, the mixture of both being complete. While this is taking place the nozzle valve remains closed; immediately upon ignition it is forced open by the pressure of the explosion, and the available energy in the gases is transformed into kinetic energy. As the pressure falls, due to the escape of the gases, the nozzle valve is forced slowly down upon its seat, its action being regulated by an oil relay, so that the air for scavenging has sufficient time to clear out the combustion chamber, and to effect a certain amount of cooling on the turbine wheel. The spring drives the valve hard upon its seat in the explosion chamber, so that during the compression of a new charge into the latter no gas can escape. When ignition has taken place, oil under pressure enters beneath the valve piston, and holds it up, keeping the valve open until scavenging is completed. Fig. 1 illustrates the first gas turbine of the type built on a working scale. It works usually with suction producer gas of 1,100 to 1,200 calories per cubic meter (123 to 134 B.T.U. per cubic foot), and has been designed for a maximum output of 1,000 horsepower at 3,000 revolutions. It drives a dynamo marked D. The turbine shaft, as will be seen, is supported from above. On it is fitted the turbine wheel and the dynamo rotor. From the very commencement smooth running was obtained at all speeds, although the temperature of the turbine casing rose to 500 degrees C. This was due to the fact that the system expands regularly and centrally with an increase in temperature. The base of the machine contains all the mechanism for the inlet and outlet of the gases and for their ignition. The material used is almost exclusively cast iron. The pressures are much less than in reciprocating engines—10, as against 30, atmospheres (147; 441 pounds per square inch); and there is no possibility of pre-ignition; the shape given the explosion chamber is a more suitable one for withstanding the force of the explosions than that of ordinary cylinders, all of which conditions combine to lead to a decrease in the thickness of the walls of the engine base (see Fig. 1). The air and gas valves are of the mushroom type; they are operated by oil relays, the oil under pressure being supplied by a rotating oil distributor. The turbine is provided with a horizontal governor shaft driven by screw gearing (see E, Fig. 1). On the shaft are keyed the main governor, an emergency governor for preventing ignition, a tachometer, the gear for operating the vertical oil distributor, and the ignition mechanism.

The action is the following:

Directly before ignition takes place there is in the explosion chamber a mixture of gas and air in the condition represented by p_0, v_0, T_0. On ignition taking place, the condition of the mixture becomes p_1, v_1, T_1, and it is presumed that when this reaction has taken place regularly, not violently, the total available heat becomes free during the explosion, and this addition of heat is made at constant volume. On reaching the p_1, v_1, T_1 condition, communication with the turbine is suddenly established. The gases are then expanded down through the nozzle to the condition p_2. v_2, T_2, and the available work they contain is transformed into kinetic energy, which in its turn is turned into work by the turbine. The variations in pressure, temperature and in the heat contained in the mixture during the process can be clearly followed in the entropy diagram, as set forth by Stodola in 1898, shortly after the appearance of the Diesel engine. The conditions in which this turbine works are, the author claims, the same as with the system of gust-governing used with the Parsons turbine. At the moment of complete combustion the explosion chamber is filled with gases,

which flow out with decreasing velocity until the pressure is that on the discharge side of the nozzle. Taking the turbine efficiency to be η, the total efficiency, he states, is

$$\eta_{tot.} = \eta \times \eta_{exp.}$$

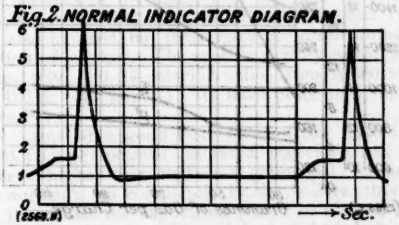
where $\eta_{exp.}$ denotes the efficiency of the explosion.

The experiments made with regard to scavenging, charging compression and regeneration may be summarized as follows:

Air scavenging is absoutey required in order to maintain the temperature of the mixture in the turbine space within reasonable limits (below 400 degrees C.). With a view to secure good economical scavenging and low heat radiation from the walls of the combustion chamber it is necessary to have the nozzle opening as large as possible. In order to save fuel this opening must be closed when charging by means of a nozzle valve. The work for air compression is obtained by utilizing the waste heat; this has no influence upon the heat balance of the gas turbine. The area of the regenerator necessary for regeneration is not greater than that of the surface condenser of a steam turbine of equal power.

The temperature of the mixture before ignition is important. On this depends the method of reaction, whether ordinary combustion or detonation; also the regularity of the process, and, above all, the explosive efficiency; it determines whether a pure mechanical process takes place, or undesirable heating. The paper includes a diagram illustrating what occurs when using carbon-monoxide gas, and states that should the temperature of the gas before ignition be 100 degrees C., then the combustion pressure will be 7.7 times the initial pressure. This will be reached in probably 3/100 to 6/100 second by a regular development of the flame. Thus the pressure of detonation will not be reached, since, starting at 100 degrees C., its ratio to the initial pressure is 9.1. But should the initial temperature be, for example, 400 degrees C., detonation will occur after quiet combustion of one-half of the charge; the remaining half will detonate spontaneously, and thereby generate a shock having 5.7 times the initial pressure, whilst the pressure corresponding to quiet burning is only 4.3 times. At 200 degrees, 14 times the value is reached as against the normal 6 times. This unexpected and not always welcome increase in pressure does not result in an increase of power, as it merely gives an unsustained shock. The paper gives a gas-turbine diagram illustrating partial ignition by explosion wave, also a diagram with normal ignition. We reproduce the latter in Fig. 2.

Fig. 2. NORMAL INDICATOR DIAGRAM.

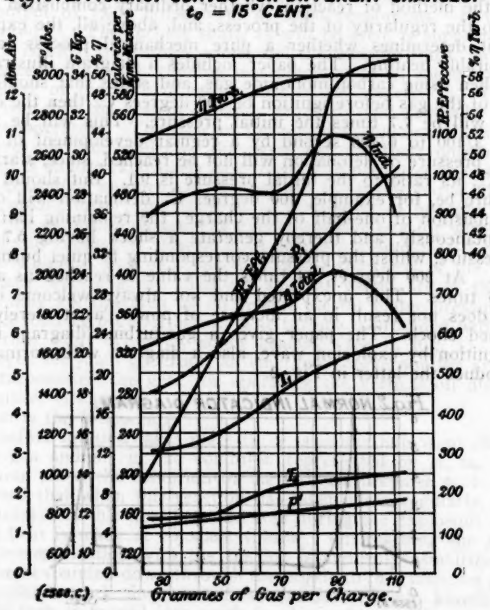


The author lays stress upon the small space occupied by the turbine as compared with that of a reciprocating gas engine of equal power. The areas occupied are as 1 to 3.28; the total weight as 1 to 4.2. The gas turbine, like the steam turbine, works without shocks or vibrations, and saves the expense of cylinder lubrication. Lubrication, as in the case of the steam turbine, is more complete and more reliable than in that of the reciprocating steam engine, since all movable parts are in oil baths. On

these grounds, the wear of the movable parts need not be expected to exceed that which obtains in steam turbines. Notwithstanding the numerous experiments made, no wear in the nozzles and blades could be found. There being absence of water particles, the causes which make for wear in the steam turbine are non-existent in the gas turbine. Starting does not require an air-compressor plant, as in reciprocating gas engines, but only a supply of steam for starting the air compressors until the regenerator supplies the necessary steam for the latter.

A first, experimental, gas turbine was built for tests in 1908. Then the one we illustrate was put in hand, and experiments with this were carried on during this present year, using producer gas from coke—*i. e.*, the gas which turned out poorest in hydrogen and in heat. The turbine, with its electric generator, ran light, excited, as early as February last, at 3,000 revolutions, 4,000 volts, with only 250 cub. m. to 300 cub. m. (8,830 cub. ft. to 10,595 cub. ft.) of gas per hour, corresponding to an indicated efficiency of over 20 per cent., "a certainly very satisfactory commencement for the efficiency curve, though only a transitory one." Then the tempera-

Fig. 3. EFFICIENCY CURVES FOR DIFFERENT CHARGES.
$t_0 = 15°$ CENT.



ture of the mixture rose largely above the value which, calculation showed, should have been the maximum reached. The total contents of the chamber soon detonated, next only a part of the charge burnt in the chamber, the other burning probably in the turbine space, not designed for that purpose. This showed how the degree of temperature before ignition influenced the ignition itself, and determined whether it occurred as combustion or developed into an explosion wave; steps were therefore taken

towards maintaining the walls of the explosion chambers as cool as possible by water, and provision was made to take the air direct through the air valves from outside. The conditions were thus improved. At the same time the method of ignition was improved, and the ignitions and combustions took place as perfectly as could be expected, allowing for the variations in the specific heat, and without explosive waves, not only for small, but also for the maximum gas charges. It is necessary, however, still to develop the engine in order actually to obtain the effect which may be expected from the calculations. With a small charge and low detonation pressure the efficiency reached was very close to that calculated. With higher charges and higher pressures, however, the increase in efficiency was not in the ratio which had been expected. The end to be aimed at in working the turbine is to obtain the curves which are shown in Fig. 3. With comparatively small charges, the end in view is already attained; further tests will lead to an equally satisfactory result, using larger charges. The curves in question, in their highest values, apply more especially to the first experimental gas turbine, and therefore not generally to the turbine we illustrate. The efficiencies given in the curves may be improved—for example, by fitting much larger nozzle exhausts, by increasing the charge pressure, the highest explosion pressure, replacing the nozzle valve by a flap, with unhindered direct through-flow of the gases, etc.

The first experimental gas turbine was built to the author's designs by Messrs. Gebr. Körting, A.-G., Hanover. The one we illustrate was built also to the author's designs by Messrs. Brown Boveri at their Mannheim Works.—"Engineering."

BOILERS FIRED WITH LIQUID FUEL.*

By Major Giulio Fumanti, of the Italian Corps of Naval Architects.

The use of liquid fuel for raising steam in the boilers of warships and merchantmen is being greatly developed. Liquid fuel may, in fact, be said to be forcing itself to the front on account of the numerous advantages which it possesses over coal, and there is no doubt that its use would be even more general than it even now is, were it not for economical conditions and for the special difficulty which exists in regard to the lack of stations for making up supplies. The liquid fuels generally burned under boilers are the residues of the distillation of crude oil at temperatures up to about 300 degrees C. (572 degrees F.). Crude oils from Texas and California can also be used, but these have the disadvantage of containing a large proportion of sulphur. The specific gravity of the liquid fuel varies between 0.88 and 0.94, and its calorific value from 10,500 to 11,000 calories (about 19,800 B.T.U. per pound). Its flash point is at about 150 degrees C. (302 degrees F.).

The calorific value of a good coal varies from 7,500 to 8,000 calories (about 14,400 B.T.U. per pound); naphtha, therefore, has a calorific value about 30 per cent. higher than that of a good coal. Seeing, further, that the coal does not burn up completely in the boilers, there occurs a loss, and it may be taken that the vaporizing power of oil and coal stand in the ratio of 15 to 10. The delivery of liquid fuel on board ship is also cheaper than coaling.

The cost of an installation, except as regards patent devices, is the same for both kinds of fuel. Apparatus is required for delivering the liquid fuel inside the furnaces; but, on the other hand, no grate, ash ejectors, and

* Abbreviated translation of a paper read in Italian before the meeting of the 1° Congresso Nazionale di Ingegneria Navale e Meccânica, held in Rome, on November 12, 1911.

so forth, are needed. The cost of working with liquid fuel is notably cheaper than that of working with coal, since the heavy expenses incurred by the frequent repairs to the grates, ash buckets, and other accessories are not incurred; the stokehold plating is less subject to wear when liquid fuel is used; the burners and the spraying installation have only to be considered in regard to handling costs, together with a higher outlay for changing the fireproof lining, which has a greater wear than in the case of boilers fired with coal; liquid fuel also requires less men in the stokeholds.

The author recapitulates the advantages of liquid-fuel firing as follows: Rapidity and facility with which it is stowed on board; reduction in the dimensions of the stokeholds, it being only necessary to provide in front of the boilers sufficient space for the combustion apparatus and for handling the burners; complete combustion and absence of smoke; facility with which the rate of combustion can be rapidly varied; the possibility of instantaneously extinguishing a furnace by simply cutting off the fuel supply; raising of steam in the briefest possible time; facility for maintaining a constant combustion in the boiler, the fuel supply being automatically constant; greater simplicity in stoking, and almost complete suppression of all fatiguing work hitherto borne by the men in the stokeholds; ease with which the fuel is stowed in the narrowest spaces, which spaces would be quite unsuitable for stowing coal. All these advantages have a very great importance, not only in the case of merchant ships, but also, and more especially, in that of warships. In the various phases which precede an engagement, when the crews have undergone a long period of cruising and navigation, they find themselves, by reason of the nervous strain to which they have been submitted, in an unsatisfactory condition from the point of view of efficiency. In moments such as these the facility with which the boilers are managed, enabling the maintenance of the speed constantly at its maximum efficiency, forms a most valuable asset. The operation in boilers using liquid fuel is the reverse of what obtains in those burning coal—the working improves as the furnace becomes incandescent; after a certain time, when all the brickwork is sufficiently hot, either the number or the supply of the burners has to be reduced in order to maintain a regular production of steam. For an equal evaporative power and, consequently, for an equal radius of action, liquid fuel leads to a saving of 40 per cent. in weight, and this equals approximately 4 per cent. of the total displacement of a battleship, 10 per cent. of that of a destroyer and 16 per cent. that of a scout. In the matter of space occupied 1 ton of coal takes up 1.25 cubic meter (43 cubic feet) and 1 ton of liquid fuel 1.08 cubic meter (38 cubic feet).

With regard to the disadvantages of liquid fuel the principal drawback is the risk of fire. To obviate this the oil is stowed in tanks located at a distance from the stokeholds and the oil is drawn from the tanks by means of tubes. Explosive gas mixtures can also form in the tanks in question when these latter are partly emptied; special care has to be taken when entering the tanks, and these, moreover, should be provided both with forced-draft ventilation and with steam-jet apparatus for expelling the inflammable gases. There may, further, occur leakages in the tubes which supply oil to the burners, and since the fuel is raised in temperature before it reaches the burners, this increases the fire risks. For this reason great attention has to be given to the manufacture of the pipe systems, and repairs must be carried out immediately the least sign of a leakage occurs. The danger is, of course, greater in the systems of sprayers in which the oil is submitted to a high pressure. One disadvantage has to be provided against when constructing the oil tanks and distributing them over the ship. The fuel being a liquid, the tanks have, for reasons of stability, to be divided up into compartments of limited capacity, and the riveting throughout has to be carefully designed and perfectly done, owing to the

facility with which the oil can leak through the joints. In warships the tanks have to be protected from gun fire; the compartments formed by the double bottom can be utilized for this purpose, and tanks can also be provided in the well, but at a distance from the boilers. When the boilers are fired with coal, this is stowed in lateral bunkers which constitute a protection; too much reliance should not, however, be placed upon this, since an engagement might take place at a time when the lateral bunkers were partly empty.

When the advantages and disadvantages of liquid fuel are considered together, the superiority of liquid fuel over coal is evident; the disadvantages, moreover, of liquid fuel can easily be overcome by a rational distribution of the tanks, and by the adoption of special methods governing the use of the fuel. The problem thus becomes an economical one; liquid fuel can be suitably adopted in every instance in which its cost is 1½ times that of coal. In warships, where the economical question is of secondary importance, and where advantages of a military character have to prevail, the preference should, without doubt, be given to liquid fuel. The unfortunate circumstance above referred to—*i. e.*, the difficulty of providing stores of oil—has, however, militated against its use. There are coal stores in all the harbors of the world, whilst petroleum stores are found in a very few places only. For this cause ships fitted exclusively for oil-fuel combustion cannot steam at any great distance from their national bases. When considering the eventuality of burning liquid fuel, two methods have been resorted to; according to one, the boilers are arranged to burn either fuel, while in the other several of the boilers are built only for liquid-fuel firing, the remainder being built for coal firing. The first method would appear to be the better of the two, the liquid fuel being kept in reserve for use only in the course of an actual engagement.

Major Fumanti then reviews in his paper the various types of burners used; he divides these into two classes, viz: (*a*) those in which the fuel is atomized by a steam or an air jet; and (*b*) those in which the liquid fuel is under a high pressure and becomes atomized by issuing from a small nozzle in the burner, this latter system being known as the mechanical-combustion system. In the first class of apparatus the oil flame is cone or fan shaped, according to the shape given the injection port; in the other, the flame is cone shaped. In the atomizers under *a*, the steam pressure is about 60 pounds to 90 pounds. The liquid fuel is, moreover, under a pressure of 10 pounds in some of the apparatus (W. N. Best and Hayes); in the Lassoe-Lovekin burner the pressure is 75 pounds. In some, the liquid fuel flows out by gravity; the Cuniberti apparatus is of this latter type. With the atomizers under *a*, the oil fuel being under pressure, a special pump is required. When gravity is resorted to the service tank supplying the atomizers requires to be placed at a suitable height above the latter. In the atomizers in which air is used the air is compressed to a pressure of 20 pounds (Gundell-Tucker); air at a lesser pressure, about 2 pounds, can also be used. In those in which the air is at a pressure of 20 pounds and over, special air-compressors are necessary; for the others, a Root or Green fan meets the purpose. Among the burners which are designed to work both with compressed air and with steam, may be mentioned the Lassoe-Lovekin burner, in which the oil is under a pressure of 75 pounds, as aforesaid, the pressure of the air or steam being about 60 pounds.

In the burners under *b* the oil is atomized under a pressure which varies from 100 pounds to 250 pounds; special pumps are required with these. The Thornycroft and Körting burners belong to this class.

In order that combustion should take place regularly the following four main conditions have to be met: Heating of the oil; filtering of the oil; proportioning the air and carefully arranging the air pipes; means being also taken for the escape of the products of combustion. The heating is necessary to give the liquid fuel the right fluidity. In the case of the me-

chanical atomizers, the oil is heated to a temperature of from 90 degrees to 125 degrees C. (194 degrees to 257 degrees F.). In that of the atomizers using a steam or an air jet, a temperature up to 60 degrees C. (140 degrees F.) is sufficient. Filtering has to be resorted to in order to prevent the small ports of the burners from becoming obstructed. The firing with oil fuel requires per pound a larger amount of air than coal. The composition of both kinds of fuel is, on an average, as follows:

Coal:—		*Oil Fuel:*	
Carbon	90	Carbon	85
Hydrogen	5	Hydrogen	14
Oxygen	5	Oxygen	1

The theoretical weight of air for burning 1 pound of coal and of oil fuel is 11.29 pounds for the former, and 14.68 pounds for the latter. In practice, however, the weight of air is double the above for the coal, and slightly under double for the oil fuel. In order to obtain satisfactory working the air supply has to be exactly that required. The Thornycroft burners are specially well suited to the warship boilers which are likely to be submitted to varied rates of working.

The flame proceeding from the oil-fuel burners extends over a great length and this condition renders it necessary to build the boilers rather longer than for coal for the products of combustion to follow a regular path without any sharp deviations. Oil-fuel firing has however been applied with success to cylindrical boilers, locomotive boilers, and also to water-tube boilers. In these latter it is necessary to arrange a system of diaphragms, according to the arrangement adopted for the tubes, in such a way that the products of combustion, after having passed through the tube sets, may have a regular flow to the chimney.

The various tests made in the Italian Navy have shown that the atomizing with a steam jet is more regular than in the other systems, since the steam heats the oil on its issuing from the atomizer, and facilitates the mixture with the air for combustion. Atomizers using an air jet have given less satisfactory results, since it would appear that the air, on issuing forth, expands, and leads to a cooling of the oil, resulting in a less good combustion. The combustion is regular with the mechanical atomizers, but at times drops of oil fall on the floor plate in front and remain unburnt; as a whole, the oil consumption per horsepower in these is greater than that with the steam-jet atomizers.

The main objection which attaches to the steam-jet atomizers is with regard to their steam or water consumption. The water consumption may be taken to amount to 0.4 pound per pound of oil. Supposing, therefore, an evaporative power of 12 pounds of water, there occurs a consumption of about 3 per cent. of the water evaporated. This point has without doubt a very great importance in the case of ships of small displacement and of great power, such as torpedo boats; and for this reason, whilst atomizing by a steam jet would be advisable for these small units, owing to their lower oil consumption, the absence of compressors, the facility in regard to repairs, etc., the mechanical atomizers have predominated for these types of light and high-speed ships. In the larger units, where the greater weight due to water and boilers has a lesser effect upon displacement, atomizing by steam or air jets has prevailed. Tests have shown that by using the steam-jet atomizers, the fans being stopped, the ship can maintain a speed of 10 knots, combustion being regular, without smoke. This cannot be done with mechanical combustion; and this is an immense advantage, since in case of damage to the fans, a ship fitted with steam-jet atomizing would not be incapacitated. Tests for visibility, carried out at night time, have shown both systems to be equally good, but with a slight advantage in favor of steam-jet evaporation, with which the rare sparks which occur with mechanical atomizing are non-existent.

With mechanical combustion it has been found that the ship is ready to start 27 minutes after the order for lighting up has been given, and 34 minutes after the same order the ship is under full steam. In 6 minutes the ship can proceed from cruising speed to full speed; 11 minutes are required to put a boiler in full steam, the others being already fired. Similar results are obtained with steam-jet atomizing; owing, however, to the system of firing, it takes longer to put the boilers in full steam. This could be remedied by adopting for ignition a small mechanical atomizer supplied by a small hand pump. Satisfactory results have obtained in Naples Dockyard with a small atomizer of this kind.

Major Fumanti then deals with the application of oil-fuel firing to merchant ships. In the instances which he gives, the oil-fuel consumption amounted to 0.500 kg. and to 0.460 kg. (1.1 pounds to 1.0 pound) per indicated horsepower per hour.—"Engineering."

CHEAPER PAINT FOR THE NAVY.

By Naval Constructor Henry Williams, U. S. Navy.

"Everywhere the paint and shellac was of excessive thickness, in places being more than one-quarter of an inch thick."—*Recent report of inspection of a naval vessel.*

Paint materials form one of the most important classes of naval supplies, and their use has contributed to many officers' reputations for efficiency through the aid they lend in making a "smart ship." There was at one time a general understanding in the British navy, reflected somewhat in ours, that no officer could aspire to the command of a first class man-of-war who, as executive officer of a similar ship, had not kept a "smart ship." This included the liberal and proper use of paint, provided often at the officer's own expense.

In these days military efficiency is the main consideration in estimating an officer's value, and paint materials cannot be considered as affecting this, other than indirectly. On the other hand, economy is being urged on all sides, and the proper and economical use of paints generally has not been given serious consideration, at least not until recently.

In order that the importance of the subject from this point of view may be judged, the value of the purchases for the year 1910 of the more important paint materials is given:

White lead	$125,000
White zinc	80,000
Red lead	50,000
Linseed oil	100,000
Japan drier	17,000
Turpentine	70,000
Shellac	135,000
Alcohol	49,000

These figures though only approximate will serve to show the relative importance from a monetary point of view of the various materials, and also justify the statement that the Navy's annual bill for painting, including the cost of labor, is not far from one million dollars.

The author has had occasion frequently to discuss the subject of paint for the Navy with paint manufacturers and chemists, men who are familiar with conditions as they exist in the trade. These gentlemen, with few exceptions, have stated their opinion that the Navy's practice in the purchase and use of paint is extravagant and not justified by the require-

ments, that equally good results could be obtained with much cheaper materials and good enough results could be obtained with still cheaper materials.

The paint and varnish industry has advanced more within the last ten years than it had advanced in the previous three centuries. It is probable that the Navy has not shared fully in the benefits due to this advance and the attitude generally adopted toward any changes in existing methods is one of conservative resistance, due frequently to lack of interest.

Ready-Mixed Paint.—Up to recently practically no use was made of ready-mixed paints in the Navy; the raw materials were issued and mixed on board ship when required for application. Early in 1911 a change was instituted by which paints mixed ready for use are issued to naval vessels in place of the raw materials. This change is in line with commercial practice, as nearly all the large manufacturing concerns and railroads issue and indeed purchase their paints ready for application. The sale of mixed paints in the United States is said to exceed sixty million gallons annually.

It is a well established and generally admitted fact that a compound paint composed of two or more pigments is far more durable than paint made of one pigment only, and that the intelligent combination of several raw materials produces the best results. Furthermore, the mixing of paint on board ship could not be done as effectively, with no paint mixing machinery or other appliances, as in the navy yards, which have power mills and mixers of the most modern types. Another advantage, and an important one, that is offered by the issue of ready-mixed paints is the facility it affords for the use of newer paint materials, which are gaining ground commercially, and which the Navy should use, if it is to avoid the deserved accusation of extravagance and of being backward in such matters. This important part of the subject will be discussed more fully in another place.

In order that a discussion of painting materials may be profitable it seems appropriate to state the characteristics and limitations of the more important of the paint materials now commonly utilized in the Navy.

White Lead.—This is the white pigment most generally used, and it enjoys a well deserved popularity among painters. For two thousand years white lead was the only white pigment used. The process by which it is manufactured most generally, known as the old Dutch process, has been in use practically unchanged for centuries, a description of it being found in Pliny's writings. It consists essentially of a slow corrosion of buckles of metallic lead in earthenware pots which contain acetic acid and tanbark. Carbonic acid is liberated from the tanbark and acting on the lead converts it into a hydrated carbonate of lead. White lead when mixed for painting purposes combines with the linseed oil, forming an unctuous substance known as "lead soap." This accounts, in a measure, for the popularity of white lead with the practical painter, who by the feel of the paint on his brush and the ease of application readily determines whether it has been made with white lead, or with "adulterants." It is a fact generally admitted that a paint made with white lead alone is not so durable as when white zinc, or even when the so-called "fillers," such as barium sulphate, silica, or similar materials are added.

For naval purposes white lead has one serious defect in its susceptibility to the action of sulphur gases. Who on board ship has not seen freshly applied white paint turn black, due to this cause? Generally for this reason the last coat of white paint to be applied to interiors of ships consists exclusively of white zinc.

White Zinc.—White zinc or zinc oxide as a paint pigment is about sixty years old, and in that time has gained a well deserved popularity. It is produced by heating metallic zinc in retorts until it vaporizes. In this condition it is passed into chambers where it oxidizes and falls as a white impalpable powder. One of the by-products is the zinc dust which is used

in the manufacture of the Norfolk Shipbottom Paint. White zinc is graded according to whiteness. The whitest is known as "French process" white zinc; that not so white is the "American process," which now is purchased almost exclusively for naval purposes. White zinc requires much more linseed oil to make a proper paint than does white lead. It dries with a hard, brittle, vitreous surface, in contrast to white lead which produces a chalky surface. Also, in further contrast, white zinc is very fine and white lead is rather coarse, so that in mixtures the interstices between the white-lead particles are filled in by those of the white zinc. Owing to the different effects of the two pigments, mixtures of them are very popular in paints, and in the Navy are generally used. Paints made with a mixture of the two pigments undoubtedly are more efficient than those made with either one alone. Similarly white zinc is mixed profitably with barium sulphate, silica, or other like substances, and paints made from these ingredients have been used recently in the Navy with good results.

Zinc oxide requires a far greater proportion of linseed oil to make paint than does white lead, and this consideration must not be lost sight of in view of the prevailing high cost of linseed oil.

Red Lead.—Red lead is the oldest of the protective paint pigments, and in this respect it is analogous to white lead. For many years no better paint for the purpose was known, and at the present time it is used very extensively and well thought of, especially among sea-going men. It is made from metallic lead by heating, first to produce litharge (PbO), and then to produce red lead or orange mineral (Pb_3O_4). When mixed with linseed oil to make paint it acts very much like plaster of paris or cement when mixed with water. The red lead gives off sufficient oxygen to harden the linseed oil and the paint becomes hard and unfit for use. For this reason no pure red-lead paints come ready for use. By the admixture of china clay, silica, or similar inert pigments, this hardening action is prevented. A very satisfactory red-lead paint is made with the following ingredients:

65 pounds dry red lead.
10 pounds dry silica.
10 pounds dry china clay.
1 pound drier.
1 pound turpentine substitute.
20 pounds linseed oil, raw.

Paint made in this way is probably more efficient than if made with red lead only and will keep without hardening.

Red lead is used on board ship for many purposes for which it is absolutely unsuited and where paint costing less than half as much would give better service. These locations are coal bunkers, double bottoms, trimming tanks and bilges. The difficulty that is encountered on the average ship in keeping the metal on these locations painted furnishes abundant proof that red lead is not suitable. It is a safe rule to follow that red lead is efficacious only for use as a priming coat, where other kinds of paint are applied over it and in locations where the metal can be rendered absolutely dry before painting. This refers especially to moisture that condenses on the metal in such spaces as double bottoms. Red lead is acted on with ease by sulphur gases when unprotected by another paint. It requires about 33 pounds of red lead to mix with a gallon of linseed oil, so that the cost per gallon of red-lead paint is something over $2.00. It is difficult to get painters to apply it properly, due to its weight and the fatigue resulting from handling the paint brush heavy with paint. For that reason there is a tendency to dilute it unduly. Some grades of red lead are coarse and crystalline, and are not so efficient as the finely divided varieties. For this reason one of the laboratory tests is an examination of red lead under the microscope in comparison with a standard sample.

Shellac.—The United States Navy is probably the largest single consumer of shellac, of which about half a million pounds are purchased annually and used largely to make varnish for linoleum-covered decks of ships. Recently considerable quantities have been used in making ship's-bottom paint.

Shellac gum, unlike other varnish gums, is not a fossil but is an annual crop. It is obtained from the secretion of innumerable small bugs which feed on the leaves of a bush growing in India and Siam. The secretion, which hardens, is called "stick-lac" and is collected by natives who take it to the factories, where it is ground, sifted, washed and winnowed and afterwards heated and spread into thin sheets to form the gum shellac of commerce. It is adulterated frequently with ordinary resin, of which large quantities are shipped to India for the purpose.

As noted above, the principal use to which shellac is put in the Navy is the manufacture of varnish for linoleum-covered decks. For this purpose it is dissolved in alcohol and a pigment, either Venetian red or yellow ochre, added to give it color and some body. On the average ship large areas of linoleum are "shellacked" each week. In spite of arguments that have been advanced that this course is necessary for sanitary reasons, there seems to be no reasonable doubt that it is mainly for appearance sake. Consideration leads to the conclusion that the cost of this luxury, which goes annually over $75,000, is hardly justified, and when the deleterious effect on the linoleum that must be exerted by the alcohol in the shellac is considered, it is a matter of wonder that the custom has continued to prevail so long. Linoleum consists of ground cork held together by a binder of oxidized linseed oil and varnish gums. A small piece of linoleum soaked in alcohol over night shows clearly the effect of the alcohol in dissolving the binder, leaving the ground cork without cohesion. The cost of providing and fitting linoleum to the decks is about $1.50 per square yard, as much as the average householder pays for his carpets. Other materials, such as ordinary floor wax, could be used for polishing the surface of the linoleum and the requirement for appearance could be met at much less cost than at present, without sacrificing to any extent the much abused consideration of "sanitary conditions."

Linseed Oil.—Linseed oil was the only oil medium for paint for many years, and is the only one at the present time used in the Navy. It is made by expressing the seeds of the flax plant, after they have been ground and heated. It has the property of combining with oxygen and, when applied in a film, of hardening into a tough covering for the surface to which applied. When heated to a temperature of 300 degrees F. to 500 degrees F., it acquires an added facility for oxidizing or drying. This is boiled linseed oil. Very much the same effect is obtained by adding so-called driers, made usually with the oxides of lead and manganese. These supply oxygen to the oil, which in turn they regain from the air. Such boiled linseed oil is called "Bunghole" oil, and is the boiled oil of commerce. Boiled linseed oil has a decided advantage for use in closed spaces on ships due to its facility of drying. It is not issued generally to ships of the Navy, the required effect being obtained through the use of driers in the paint.

Due to an unprecedented condition in the linseed-oil market the price has tripled in recent years. The paint industry is the only one that adheres to the use of linseed oil, although there are substitutes that promise well and undoubtedly can be developed to take its place in part if not entirely. The most important of these substitutes is fish oil, which is made from menhaden and now can be produced without the disagreeable odor that marked it formerly, at about half the price of linseed oil. Experiments made with fish oil for paints for naval purposes promise well and its use to a certain extent is likely.

There is no justification for the disposition of many to consider fish oil or similar materials as adulterants, and many manufacturers of oil cloths,

patent leathers and imitation rubber fabrics have found that, when properly treated, it is superior to linseed oil. It can be treated so as to withstand the action of high heat, and for this reason is valuable for smokestack paint. Dr. Maximilian Toch is authority for the statement that the addition of fish oil with suitable drier renders paint more durable to sea air and corrects the chalking tendency of white lead and the hard drying quality of white zinc.

It is probable that fish oil and other oils are destined to supplant, to a considerable extent, linseed oil for painting, as the conditions that have led up to the scarcity and consequent high price of flax seed are not likely to be relieved.

China wood oil, or Tung oil as it is called also, can not be regarded as a substitute for linseed oil except for certain special uses. Linseed-oil films have not the property of excluding moisture and for that reason linseed-oil paints will not stand well in damp locations. Paints made with Tung oil have the quality of being waterproof and its use in damp-proof paints is general for that reason. Tung oil is difficult of manipulation and to obtain good results considerable skill is necessary. In spite of this, however, its use is becoming general, not only in paint above mentioned, but also in varnishes. Dr. Toch is authority for the statement that a better varnish can be made out of Tung oil and resin than out of higher-grade gums and linseed oil.

Soya-bean oil and corn oil are used for mixing with linseed oil. It has been stated on good authority that as much as 20 per cent. of them can be mixed with linseed oil without affecting the properties of the latter, except as to drying.

Turpentine.—Turpentine has been used for many generations as a solvent for paints and varnishes. It is manufactured from the sap of several species of pine trees, by distilling with steam. Due to the destructive methods employed in gathering turpentine the supply has decreased enormously in recent years, and the price has increased correspondingly. Unless these methods are interfered with, it is likely that in the lifetime of the present generation turpentine will become a chemical curiosity. It is used to mix with paints as a diluent or extender, its function is mechanical, being simply to increase the spreading power and working quality of the paint. It evaporates leaving no residue, so it cannot be said that it contributes except indirectly to the life or quality of the paint film. It is generally admitted now that entirely too much stress is laid upon the value of turpentine as a paint vehicle, and when this is realized substitutes will be used that serve the purpose quite as well and in many cases better than turpentine itself.

Due to the scarcity of sap turpentine, of recent years so-called wood turpentine has been manufactured extensively. This is made by the distillation in retorts of sawdust, shavings, old logs and roots of pine trees. The quality of these turpentines depends upon the method and care used in their manufacture, many of them being the equals in every respect, except possibly as to odor, of the sap turpentine. Others have very disagreeable odors and injurious properties due to the products formed in the destructive distillation of the wood. For exterior painting these wood turpentines generally are quite as efficient as the sap turpentine. For interior painting their odor in most cases condemns them. In general they cannot be said to have any advantage over benzine products, which are supplied for the same purposes.

The Navy Department purchased in the year 1910 about 75,000 gallons of turpentine under specifications that required delivery of the highest grade of sap turpentine that it is possible to obtain, and consequently at prices somewhat higher than the market. This turpentine was used by the Navy for paints of the sort in which manufacturers of ready-mixed paints would not consider using other than benzine products as solvents. Consideration of these facts led to the decision early in 1911 to

discontinue the purchase of sap turpentines except for special purposes and to purchase for general use in paints a turpentine substitute which had been shown by test to meet all practical requirements. This material is a petroleum product similar to benzine, but with higher flash point. It evaporates at a temperature of 212 degrees F. with no residue. The use of the material has met with some opposition in the Navy, reports have been submitted as to its unsatisfactory qualities. It should be borne in mind, however, that it is a commercial product used very extensively commercially for the same purposes as the Navy uses it, and any failure, consequently, should be considered rather as the fault of the persons using it than of the material. It is likely also that failures due to other causes are attributed to the use of the turpentine substitute. A conservative estimate of the saving to the Navy resulting from the use of turpentine substitute is $30,000 annually. One of the objections advanced frequently and with effect against petroleum thinners is the belief that they have a tendency to cause white paints to turn yellow. That this is true is doubtful. The author has never been able to understand that very white paint should be necessary on board ship, and in discussing the matter with expert house decorators finds that from artistic considerations they favor a cream or light yellow rather than dead white. This tint permits the use of more suitable varnish in the gloss paint, as will be pointed out further along.

Iron Oxide Paints.—Large quantities of venetian red are purchased in the Navy for coloring the shellac varnish used on linoleum decks. It consists largely of calcium sulphate or gypsum, the remainder being about 30 per cent. of iron oxide. One of the most durable of the proprietary metal paints tested at the Brooklyn Navy Yard has a pigment consisting of venetian red.

Indian red is a natural oxide of iron, practically pure, and is used as a coloring matter in the Norfolk anti-fouling ship-bottom paint, in order that it may be distinguished readily from the anti-corrosive.

Metallic brown is another oxide of iron—the best grades being the natural ones. It is used very extensively commercially on steel with success, and is considered to be one of the standard protective paints, many persons thinking it the equal of red lead for this purpose.

Black Paints.—There are many forms of black paint pigments, all of them practically pure carbon, though they differ in structure and qualities. The one most commonly used in the Navy is lamp-black, which is made by the combustion of oils. It has remarkable tinting strength and the outside paint now used on ships owes its slate color to the addition of lamp-black, the base being of practically the same constituents as the white paint used formerly.

There are a number of graphite pigments, all of them being nearly pure carbon; alone they form an unsatisfactory paint pigment but added to other heavier bases graphite pigments have great merit.

Red Paint.—For many years quantities of English vermilion paint were used on naval vessels for painting the inside of ventilator cowls. There is no doubt that this added considerably to the smart appearance of the ships in contrast to the white hulls and salmon-colored top sides.

English vermilion is a sulphide of mercury and has the defect of hardening when mixed with oil and kept. The purchase of English vermilion for the Navy was abandoned a number of years ago, in line with commercial practice, in favor of the so-called artificial vermilion. This is cheaper, less likely to change color and can be kept mixed ready for use. The particular pigment used in the Navy is a lead-barium lake of the azo dye, known commercially as "Lithol."

Red, as well as green, yellow and blue paints, do not have the same importance in the Navy as was the case formerly, due to the adoption of the slate color for exteriors of ships. Their use is confined almost exclusively to painting the distinguishing striping on piping on board ships.

Bitumastic Paints.—Specifications for recent battleships require that the builders, in lieu of the red lead and other paints usually required for these localities, shall apply to coal bunkers, trimming tanks, reserve-feed water tanks, inner bottoms in machinery spaces, and bulkheads and foundations in machinery spaces to height of floor plates, the so-called bitumastic paints or their equivalents. These materials are supplied by several firms under different names, all of them being of similar composition and characteristics. A discussion of them seems proper, as they promise to play an important part in future painting of the spaces mentioned above, which in the past on naval vessels have been a source of considerable worry to the officer responsible for their up-keep. Coal-tar products deteriorate when exposed to light, and for that reason these materials can be applied only in confined spaces. Formulas are given for their manufacture which have been determined tentatively and, while they should not be regarded as final, they will serve to indicate the character and general ingredients of the materials.

The first coat to be applied in all cases is the so-called "solution," which may be made by melting 100 pounds of coal-tar pitch and 50 pounds of Trinidad asphalt. When liquid the mixture is removed from the fire and cooled and thinners added consisting of 65 pounds of coal-tar naphtha and 15 pounds of mineral oil.

The "enamel" may be made by melting 350 pounds of coal-tar pitch and 350 pounds of Trinidad asphalt together and boiling for about three hours.

The "cement" may be made by melting 150 pounds of coal-tar pitch and 120 pounds of Trinidad asphalt together and boiling for about three hours, when 150 pounds of Portland cement is added and stirred in.

In all of these care is required in the heating; the quality of the material is dependent entirely on the manner in which this is accomplished.

The "solution" is applied first in all cases and does not require to be heated, but is brushed on cold. In some spaces the other materials are unnecessary, though it is best generally to follow up the solution with the enamel or cement, both of which must be heated to apply. The former goes on in thickness of about 1/16 inch, the latter about ½ inch. The cement is applied only to the bilges in the machinery spaces, and in the firerooms frequently it is protected by a covering of Portland cement.

These materials have been tried to a limited extent on naval vessels, but have been used extensively on merchant ships for many years. The author inspected, in 1904, the American Line *St. Louis*, which had had the bitumastic materials applied at the time of building in 1894-1895 to trimming tanks, chain lockers, double bottoms, coal bunkers and inner bottoms. The original material was in good condition and had received only minor repairs. The metal underneath generally showed no signs of corrosion and in many cases the hammer marks around rivet heads could be seen when the bitumastic was removed.

The So-called "Inert Fillers."—For many years white lead, white zinc, linseed oil and turpentine have been regarded as the standard ingredients of which any paint should be made. Many manufacturers quietly utilized other materials that gave results quite as good, but the consuming public cared mainly about the results and had no means of determining the ingredients. A general tendency grew up to regard as adulterants many other pigments than the above that have excellent qualities to recommend them as paint ingredients. Among these may be mentioned lithopone, blanc fixe, barytes, whiting, gypsum, china clay and silica. Some years ago a number of States passed the "pure paint laws" that required, in effect, labelling packages of paint with a statement of the contents. This precipitated the situation, in which many manufacturers find themselves, of being forced to justify their use of materials that have given satisfactory results, but which they have used unheralded and which they have permitted the public to be taught to consider as adulterants. An effective campaign now is being waged to show that all of these materials have

their function when used in the proper proportions in paints, though their unskilled use perhaps may not give as satisfactory results as older and better known pigments. It was demonstrated by actual exposure tests at the seashore that paint in which the much-despised barytes was a principal pigment gave as good results as one made up with only white lead and white zinc. Many of the numerous kinds of paints used by the agricultural implement trade are made largely from barytes and the results obtained from these should be a sufficient lesson to the Navy that its paint money could be spent to better advantage than in the exclusive purchase of lead and zinc paints.

Lithopone is a remarkably fine white pigment consisting of about 70 per cent. of barium sulphate and 30 per cent. of zinc sulphide. It has achieved great vogue recently for use in the flat-wall finishes, which promise to supplant wall paper to a large extent, due to their sanitary properties and durability. It is not suitable, however, for outside paint.

Blanc fixe is barium sulphate artificially produced and is superior to barytes in its fineness and texture. It has many qualities to recommend it for naval purposes, and experiments looking to its use have been undertaken, with excellent practical results. Slate-color paint for outside of ships made according to the following formula has given excellent results in preliminary tests:

450	pounds blanc fixe, dry.
70	pounds French ochre, dry.
467	pounds white zinc, dry.
93	pounds graphite in oil.
3	pounds lamp black in oil.
93	gallons raw linseed oil.
12	gallons oil drier.
8¾	gallons turpentine substitute.
2	gallons benzine.

Barytes is well known as the white mineral barium sulphate. Used alone it has little covering power, but when mixed with white zinc or white lead it gives excellent results. Consideration shows that it would not be unreasonable to expect it to show durability as a paint pigment, as it is one of the most permanent of the minerals and is absolutely insoluble in most solvents. Due to its density and crystalline structure, barytes has not large covering or hiding power, and needs to be mixed with other pigments to form satisfactory paint.

Whiting, or calcium carbonate, is used mainly for making putty, but small percentages are used by many manufacturers in all paints for the purpose of neutralizing any acidity that may develop. This feature of whiting recommends it highly for paints for steel surfaces.

Gypsum, or calcium sulphate, is the main ingredient of venetian red, of which large quantities are used by the Navy, the remainder being ferric oxide. In its pure state it is used to advantage if free from lime.

China clay is used mainly as a filler for paper, but has considerable value in paint mixtures, its addition being an improvement to red-lead paint as seen above.

Silica is useful in mixing with other pigments, largely because of its quality of presenting an excellent surface for repainting. This quality alone should recommend it for use in many of the paints used on shipboard, and the statement has been made that the time is not far distant when all good mixed paints will contain silica.

Asbestine is a silicate of magnesium, and is fibrous in character. It is extremely stable and when used in mixed paints has a tendency to hold up the heavier pigments and to keep them from settling to the bottom of the container. It is used considerably in ready-mixed paints for this reason.

Varnishes.—Varnishes are made ordinarily by melting the gums or resins, then adding hot linseed oil and, when the mixture has cooled sufficiently, thinning with turpentine or petroleum spirits. The varnishes intended for outside use, usually known as "spar varnishes," are made with better and harder gums and do not contain so much volatile solvent and consequently dry more slowly. Varnishes for interior use are usually not so expensive, have more volatile solvent and dry much more rapidly even in confined spaces.

Considerable skill is required in the manufacture of varnish and the results vary considerably between the products of different makers, even when the same materials have been used. Some makers, by the use of lower-grade gums and even common pine resin, can produce better varnishes than their competitors can with the best materials. The only test of varnish that can be relied on is a service test, which, however, is prohibited by the length of time required before the varnish will wear out. It is, therefore, necessary to purchase varnish on specifications which require a chemical test only, and, though this cannot be regarded as showing conclusively the quality of a varnish, it is fair generally to assume that varnishes in which only the best materials have been used are of good quality and will give satisfactory results.

Damar varnish, which is made from damar gum and turpentine, without the use of oil, is used in mixing enamel paints for interiors of living spaces on ship-board. It was selected for this purpose due to its light color, although it is not so durable as the linseed-oil varnishes. With a slight sacrifice of whiteness, a gloss paint could be made by using an oil varnish that would be much more durable than damar-varnish paint and cost no more. The flat-wall finishes made from lithopone that have become popular recently, contain a small percentage of varnish and for that reason will stand scrubbing quite as well as gloss paint made with damar varnish.

Asphaltum varnish is made in the same way as other oil varnishes by using a fossil asphalt, known as gilsonite, with linseed oil and a solvent. This kind of varnish has not very great durability.

Varnish gums or resins are fossil remains of exudations from trees and there are many varieties and grades. The older gums are harder, amber being the final result of aging. Varnish has been made with amber, but, due to the extreme heat necessary to melt it, the resulting varnish was dark colored and had no particular merit.

Special Rust-Preventing Paints.—The subject of corrosion of steel has received of late years considerable attention commercially, due to the amount of money being invested in buildings which depend for their existence upon the integrity of their steel structures. This steel is not subjected to the trying conditions that a sea-going ship experiences, for the reason that it is imbedded usually in cement concrete, and, morever, is not exposed to salt water or air laden with salt moisture.

As a result of these efforts there have been developed a number of paints designed especially to prevent rusting of steel surfaces. While it would not be possible to discuss all of the different kinds and characters of these paints, it is believed that description of several of them, that have been used successfully to some extent in the Navy, will be of interest in showing the possibilities and the breadth of the field.

A paint that has been used extensively in steel work of recent buildings contains as a pigment a mixture of oxides of silica, calcium and aluminum, resembling closely Portland cement, which is mixed with other materials and china-wood oil. The latter has the property of forming a moisture-excluding film, while the cement absorbs, when applied, moisture that may be present on the surface and permits the use of such a paint on moist surfaces. These latter characteristics are sufficient to justify the substitution of such a paint for many localities in which red lead now is used. Its price, which is one-third less than that of red lead, and its protective and rust-preventing qualities, which have been shown by experiment to

equal if not surpass red lead, are added reasons in favor of the adoption of such a paint for naval vessels.

A paint that has given excellent results when used on the interiors of trimming tanks of submarines is made with a graphite pigment and an asphaltic oil thinned with benzine. This paint has displayed remarkable protective qualities when used under the trying conditions mentioned above and its cost is less than one dollar a gallon. Red lead, which costs not far from two dollars a gallon, though used generally for the purposes mentioned, does not give results that are even reasonably satisfactory, due to the fact that linseed-oil films do not exclude water, and red-lead paint made with linseed oil will not adhere properly to a surface that has the slightest trace of moisture.

Another paint that has been tested extensively at the Brooklyn Navy Yard, for uses to which red lead has been put, is composed of a pigment similar to venetian red, containing iron oxide, calcium sulphate and silica, and of specially high quality of linseed oil, turpentine and driers. The merit of this paint appears to be due to the special quality of the linseed oil and the care used in manufacture, and, though it is somewhat more expensive than the others mentioned, the paint is much cheaper than red lead and appears to be more effective.

Conclusion.—An important consideration, and one that should not be lost sight of in connection with the paint situation in the Navy, is the fact that rarely is paint allowed to live its full life, or the paint film to perish through the action of the elements. Occasionally in inaccessible places on ship-board the metal becomes badly corroded, either under paint, or else due to its absence. These cases are unimportant and can be attributed generally to the unsuitability of the paint used. Most of the paint, however, that is used on ships is painted over before it has perished. Ample proof of this is the thickness of the paint on various parts of old ships, and the scaling of old paint that goes on continually on men-of-war. Granting, for the sake of argument, that the new paint materials, such as turpentine substitutes, linseed-oil substitutes, the inert fillers, etc., have not the durability of the materials heretofore used, it would even then be reasonable to use them. When a house on shore is painted, the owner expects the paint to last at least two years and would not consider cheaper materials if he felt that it would shorten the life of the paint appreciably. The labor cost of painting is too high. He does not expect to repaint until the paint film has disintegrated and the wood or metal begins to show through. Ships are repainted, however, for other reasons, and it is rare that any part of the ship that is visible goes more than one year without repainting, most of it being painted three or four times a year.—"U. S. Naval Institute."

PUMPS.

By a stretch of the imagination the following, which is an authentic letter written by a Cadian of Louisiana, may be considered to treat of an engineering subject:

"La., 1911.

"Mr. ———,

"Dear Fren.—

"i get the pump which you by for me, but why for g— saks you don send me no handle. Wats the use a pump when she don have no handle. i loose to me nu customer. Shure thing you don tret me rite. i rote ten day and me customer he holler for water like h— for the pump. You know he is hot summer now and the win he no blow the pump. She got no handle so what the h— i goan to do with it. Doan send me the handle

pretty quick, i send her back and i goan order some pump from woodwight companie new orleen.

"Goodbye,

"Yours truly,

"———— ——— ————."

"Since i rite i find the g— d— handle in the box excuse to me."

DRAFT AND DIFFERENTIAL GAGES.

By Julian C. Smallwood.

There are three units commonly used in the measurement of fluid pressure: pounds per square inch, inches of mercury, and inches or feet of water. Steam pressure is measured in pounds per square inch, back pressure or vacuum in inches of mercury, draft in inches of water, and water pressures in pounds per square inch or in feet of head. As a rule the pressure of the atmosphere (14.7 pounds per square inch) is selected as the point from which measurements are made, the ordinary pressure gage reading so many pounds above atmospheric pressure, the vacuum gage so many inches of mercury below it, and the draft gage inches of water above or below the atmosphere. In some cases merely the difference between two pressures is required, for which purpose a differential gage is used.

The conception of pressure in pounds to the square inch needs no comment. A cubic inch of water at 60 degrees F. weighs about 0.036 pound; hence a column of water 1 inch high and 1 square inch in cross-section will exert a pressure by its weight of 0.036 pound distributed over the square inch of its bottom surface of contact. If the column is made 2 inches high, this pressure is doubled, and so on.

Mercury at 60 degrees F. weights nearly 13.6 times as much as water; therefore each inch of mercury exerts a pressure of—

$$13.6 \times 0.036 = 0.49 \text{ pound per square inch.}$$

The simplest form of draft gage, the U-tube, is that shown in Fig. 1. This is a piece of glass tube bent in the form of a U and partly filled with water. It is connected to the flue at A by a rubber tube, and when the two water levels are the same the pressure in the flue is the same as that without. But if the level on the flue side of the U-tube is H inches higher than the other, then the pressure of the atmosphere on the open side of the tube is enough greater than the flue pressure to raise the weight of H inches of water against it. From this difference in level the difference in pressure is obtained between that in the flue and atmospheric pressure expressed in inches of water.

Such an instrument is easily made, and can be put together at small cost. It is not always easy to bend the glass into the form of a U, in which case two straight pieces of glass may be joined by a piece of rubber tubing, as shown in Fig. 2. These may be mounted on a board to which a paper scale is glued, and afterwards varnished.

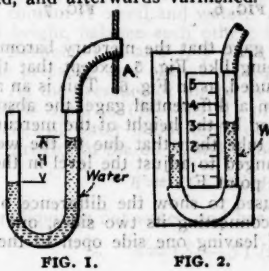


FIG. 1. FIG. 2. FIGS. 3 and 4.

The simple U-tube is applicable also to pressures greater as well as less than atmospheric, the high level then being on the open side. For measuring pressures of more than one pound, however, the water gage becomes inconveniently long. Hence for this purpose a heavier liquid, such as mercury, which will not rise so high, is used. Each inch of mercury being equivalent to 0.49 pound per square inch, indicates a pressure 13.6 times as great as that shown by water.

The scales of these gages are graduated in various ways. Some have an ordinary scale, as in Fig. 2, which requires the height of both levels to be read, and then the lower subtracted from the upper to obtain the difference in level. Others are graduated as shown by Fig. 3, the purpose being to avoid two readings. A U-tube supplied with this scale must be filled until the level on both sides is at the zero graduation. Either the upper or the lower scale is read, and the reading is doubled to obtain the inches of water. If any of the water is evaporated or otherwise accidentally lost, however, the reading will be inaccurate.

Another form of scale is shown by Fig. 4, which obviates doubling the reading. This may be made adjustable through a small range, so that its zero may be readily brought to coincide with the no-pressure position of the water level.

A gage giving direct readings may be made as shown in Fig. 5. This consists of a cup C of comparatively large diameter containing liquid, in which a glass tube of small bore is placed. As the liquid rises in the tube the level in the cup sinks, but on account of the large area of the liquid in the cup its level does not sink appreciably. Thus, if the cup is 2 inches in diameter and the tube ⅛ inch bore, for each inch the level in the tube rises, that in the cup falls 1/256 inch. If the gage range is 5 inches, the total fall of level in the cup is about 1/50 inch, which in this case is negligible. If the cup is a closed chamber, except for a tube leading into its air space, the instrument may be used as a differential gage.

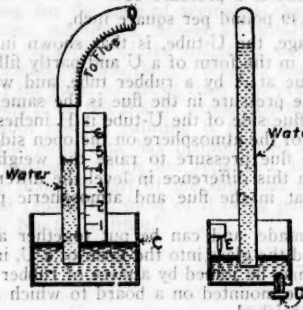


FIG. 5.

FIG. 6.

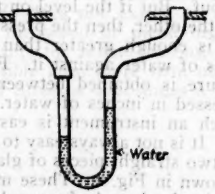


FIG. 7.

It is on the principle of this gage that the mercury barometer is made, the form of the instrument being like Fig. 5, except that the glass tube is closed at the top and air excluded, as in Fig. 6. This is an absolute-pressure gage as distinguished from a differential gage, the absolute pressure of the atmosphere being recorded by the height of the mercury column, as there is no other force in the tube than that due to the weight of mercury. A thumbscrew D is arranged to adjust the level in the open vessel at a fixed height shown by the point E.

The simple U-tube may be used to show the difference of pressure in two parts of a closed pipe by connecting its two sides, one to each part, as shown by Fig. 7, instead of leaving one side open to the atmosphere.

When measuring forced draft it should be connected in this way: one side to the ashpit, the other to the chimney flue. The pressure in the ashpit is greater than that of the atmosphere; that above the fuel bed may be less.

In some appliances for measuring the flow of steam the U-tube may be used, its form being somewhat different for practical purposes. When the mercury is employed, water of condensation settles on one side of the mercury column and influences the reading. This may be corrected as follows: Considering Fig. 8, the true difference of pressure is—

$$\frac{H + H'}{13.6}$$

inches of mercury; that is, the water column in inches is divided by 13.6 to obtain the additional pressure in inches of mercury. If the water were on the other side it would be subtracted.

As applied to furnace drafts, the U-tube as described is not satisfactory, because ordinarily, with natural draft, the difference does not exceed a few tenths of an inch. Therefore the instrument is difficult to read closely, and does not indicate small changes of draft. To overcome this difficulty a great many ingenious devices have been constructed.

It would appear at first that the difficulty could be overcome by employing a liquid lighter than water. It is not practicable, however, to obtain one enough lighter to magnify the difference of level sufficiently, but two liquids, differing from each other in density, will answer the purpose. Thus in Fig. 9 assume the upper liquid to be oil whose specific gravity is

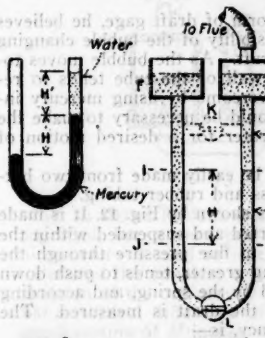


FIG. 8.

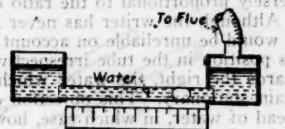


FIG. 10.

FIG. 11.

FIG. 9.

0.9, and the lower liquid water. If the levels in the cups F, F are the same, then there is a difference of pressure corresponding to the difference in the weights of the column H of oil and water. Above I, I, the columns of oil, being of equal height, balance each other; below J, J the water columns also balance. Between I, I and J, J there is a column of water acting against the flue pressure, and one of oil, equal in height, acting with the flue pressure. As the pressure of the oil is 0.9 that of the same height of water, the draft is equal to (H — 0.9 H) or 0.1 H inch of water; that is, the draft in inches of water is one-tenth of the reading. In general, calling the specific gravity of the oil S, and h the draft in inches of water—

$$h = (1 - S)\, H.$$

If the heavier liquid is not water, and has a specific gravity, S_1, then—

$$h = (S_1 - S)\, H.$$

By choosing the liquids for such a gage of nearly the same densities, any desired range may be obtained for the measurement of a small difference of pressure. The more nearly equal in weight they are, the greater the motion of the liquid for a given draft. In general, the number of inches of rise in the ordinary U-tube is multiplied by $\frac{1}{S_1 - S}$.

In practice, when used as a draft gage, the difference of level in the cups F, F would alter the foregoing relation. To keep the cup levels the same a cross-connection with a cock K joins the tops of the tubes. Another cock L is placed at the bottom of the tubes. When in use K is closed first and L opened; when the oil stops rising in the right-hand tube, L is shut and K opened, and the flue connection is removed. This allows the oil in the cups to come to the same level again without disturbing the levels in the tubes. The cocks are then reversed until the oil has risen again as high as it will, and the level in the cups is adjusted once more. This is repeated until the oil no longer rises, upon which the reading is taken. This is the principle of Hoadley's draft gage.

The draft gage shown in Fig. 10 consists of two cups, one closed, the other open to the atmosphere. They are connected below the water line by a horizontal tube of small bore containing water in which there is a small air bubble. The motion of this bubble is a measure of the draft. As the water rises in the right-hand cup the bubble moves to the right a distance proportional to the difference of level in the cups. By making the ratio of cross-sectional areas of the cup and the tube bore the required amount, any desired range of motion may be obtained, the motion being inversely proportional to the ratio of areas.

Although the writer has never used this form of draft gage, he believes it would be unreliable on account of the possibility of the bubble changing its position in the tube irrespective of the draft. As the bubble moves towards the right, the water which wets the wall of the tube tends to remain stationary. This objection could be overcome by using mercury instead of water, in which case, however, it would be necessary to make the ratio of cross-sectional areas very much greater for a desired motion of the bubble.

Fig. 11 shows how this form of gage may be easily made from two bottles with perforated corks, together with glass and rubber tubing.

A draft gage designed by Professor Kent is shown by Fig. 12. It is made of two tin vessels M and N, the former inverted and suspended within the other by a spring. Its interior is subjected to flue pressure through the tube O. The pressure of the atmosphere being greater, tends to push down the suspended can. This tendency is opposed by the spring, and according to its extension recorded by the pointer P, the draft is measured. The equation of the instrument, neglecting buoyancy, is—

$$p \times a = 0.036\ h \times a = n \times H,$$

from which—

$$h = \frac{n \times H}{0.036 \times a} = k \times H;$$

where—

p = Difference between flue and atmospheric pressures in pounds per square inch;
a = Internal cross-section, in square inches, of can M;
n = Number of pounds corresponding to 1 inch extension of spring;
H = Extension of spring in inches corresponding to $p \times a$ pounds;
h = Draft in inches of water;

$$k = \frac{n}{0.036\ a}.$$

The equation ($h = k\ H$) shows that to obtain the draft the reading on the scale is multiplied by a constant which is less than one. The increasing buoyancy as M sinks is negligible if the design of the gage is good. This instrument does not quickly respond to changes of draft, however, and is cumbersome.

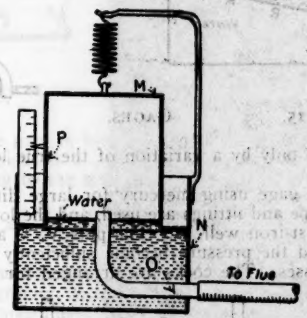


FIG. 12.

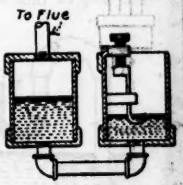


FIG. 13.

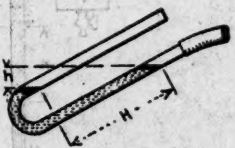


FIG. 14.

Fig. 13 shows Miller's draft gage, in which the difference of level is not magnified, but a very precise instrument—the hook gage attached to a micrometer—is used to measure the level.

Perhaps the simplest and most satisfactory form of draft gage is the variation of the ordinary U-tube shown by Fig. 14. It is set on a slant, so that a vertical difference of h inches between the water levels causes an actual motion of H inches. By varying the slant any desired magnification may be obtained. This is generally made with a scale on the principle illustrated in Fig. 4, so that only one reading need be taken. It may be readily constructed of a piece of gage glass Q (Fig. 15) and a glass tube R, together with some rubber tubing; perforated corks S may be used if forced draft is to be measured. The gage should be mounted on a board to which a spirit level T is attached, so that the slant of R may be kept as intended.

For the purpose of making this gage more compact the slanting tube is sometimes in the form of a helix.

Most of the previously described gages are applicable to the measurement of water as well as gas pressure, the liquid used in them for this purpose being mercury or oil. Fig. 16, for example, shows the ordinary U-tube inverted for use with oil. The interpretation of the reading is the same as explained for the Hoadley gage. It is important to exclude air bubbles from the piping system of such a gage, and this can be done by carefully introducing the oil or mercury through cocks. A column of air in the connections on either side would cause a reduction of weight which

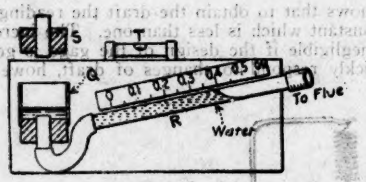


FIG. 15.

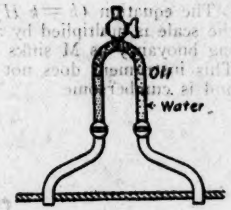


FIG. 16.

GAGES.

would be balanced only by a variation of the true level difference of the gaging liquid.

Fig. 17 shows a gage using mercury for large differences of pressure. One-eighth-inch pipe and fittings are used, and the long gage glasses U, U are joined by a cast-iron well W. The pots V, V are arranged to catch the mercury should the pressure become sufficiently great to blow it beyond the gage glasses. The cocks are arranged for introducing the mer-

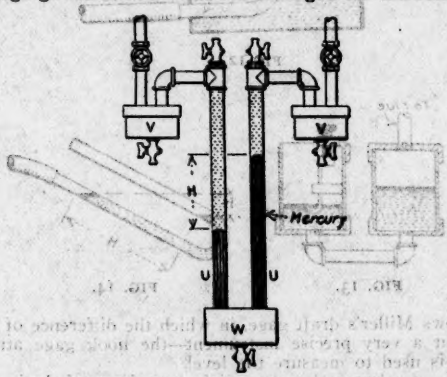


FIG. 17.

cury and excluding the air. In the right-hand column there is acting a weight of H inches of mercury which is partly balanced by the same height of water on the other side. The mercury column causes a pressure of 13.6 H inches of water; therefore the pressure difference indicated by the gage is—

$$13.6\ H - H = 12.6\ H \text{ inches of water,}$$

or 1.05 H feet. That is, to interpret the reading of this gage its reading in inches must be multiplied by 1.05 to get the difference of pressure head in feet.

In Fig 18 the symbols P, V, and A stand for pressure head in feet, velocity, and area in square feet, respectively, at the two sections of the pipe. Since the sections at A_1 and A_2 are different, the velocities will be different, V_2 being as many times V_1 as A_1 is times A_2. Neglecting the losses due to friction, etc., the total energy of the water in the one section is equal to that in the other, this energy being in two forms—pressure and velocity. Since the velocity in the small pipe is greater than that in the large, the pressure in the latter must be less and the diminution in pressure

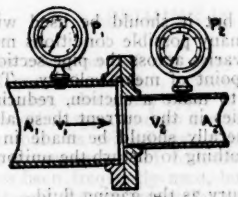


FIG. 18. GAGES.

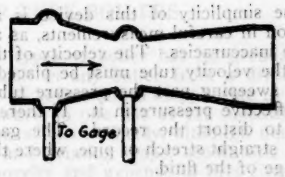


FIG. 19.

energy must equal the gain in kinetic energy. Therefore the difference of pressure in the two pipes is a measure of the velocity, and consequently the quantity, according to the following relation:

$$Q = \frac{A_1 \times A_2}{\sqrt{A_1^2 - A_2^2}} \sqrt{2\,g\,(P_1 - P_2)},$$

in which Q is the quantity flowing in cubic feet per second and g is the acceleration due to gravity, 32.16. This is the principle of the Venturi meter, a very precise instrument for the measurement of the flow of water. It has also been applied to steam and perfect gases, but with less success. Fig. 19 shows the usual proportions of the meter, the actual discharge being between 95 and 99 per cent. of that shown by the formula.

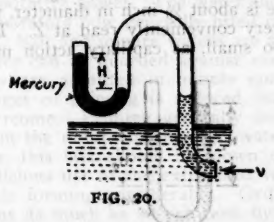


FIG. 20.

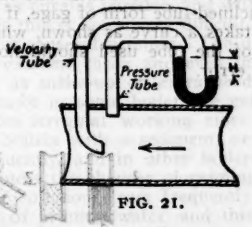


FIG. 21.

Fig. 20 represents a Pitot tube, another instrument for measuring velocities and quantities. It is a tube having a quarter bend, set facing the direction of flow. The height H of the liquid in the tube is a measure of the velocity, or—

$$V = \sqrt{2\,g\,h},$$

h being the head in feet of whatever liquid or gas is flowing. To convert the reading H to the equivalent head of the fluid whose velocity is being measured, it is necessary to multiply by the ratio of the densities of the gaging and the measured fluids. Thus, if the gaging liquid were 10 times as heavy as the fluid being measured, h would equal 10 H. In general—

$$V = \sqrt{2\,g\,C\,H},$$

in which C is the ratio of densities.

The Pitot tube has been used largely for velocity measurements in open streams, but not until comparatively recent years has it been applied extensively to the flow of liquids and gases in closed pipes. Fig. 21 shows its arrangement for this purpose. Since in such a system pressure is recorded as well as velocity, a pressure tube is added. Static pressure only acts through this tube and balances the static pressure in the other tube; hence the gaging liquid records the velocity head only.

The simplicity of this device is alluring, but it should be used with caution in careful measurements, as a great many possible conditions may cause inaccuracies. The velocity of the fluid varies across the pipe section, and the velocity tube must be placed at the point of mean velocity. The fluid sweeping past the pressure tube tends to make a suction, reducing the effective pressure in it. If there are eddies in the current these also tend to distort the record. The gaging generally should be made in a clear, straight stretch of pipe, where there is nothing to disturb the uniform passage of the fluid.

When used for the flow of water with mercury as the gaging fluid—

$$V = 8.23 \sqrt{H},$$

and for the flow of air at atmospheric pressure and 60 degrees F., water being the gaging fluid—

$$V = 66.3 \sqrt{H},$$

in which V is the velocity in feet per second, and H the reading in inches

The gages described are useful in measuring losses of pressure in various parts of blower systems. By their use leaks may often be located and otherwise unsuspected losses remedied.

Practical Hints.—The level of mercury in a glass tube curves from the center downward, while that of water curves upward. It is best to read the height on a horizontal line tangent to the curve, as Z, Z, Fig. 22. In the inclined-tube form of gage, if the tube is about ⅛ inch in diameter, the level takes a curve as shown, which is very conveniently read at Z. The bore of the tube used should not be too small, as capillary action may cause error.

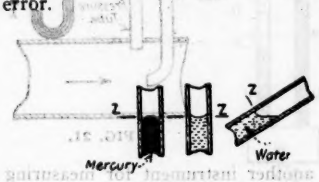


FIG. 22.

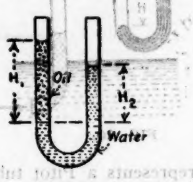


FIG. 23.

When oil is used as a gaging fluid its specific gravity may be found, as shown by Fig. 23. A little oil is floated on water in one side of a U-tube, and its height H_1 and that of the column of water balancing it H_2 are measured. Then the specific gravity of the oil is $\frac{H_2}{H_1}$.

Considering the units of pressure measurement dealt with, it is unfortunate and illogical that three different physical quantities and two starting points for measuring are employed when one would do. There is no reason why the ordinary spring-pressure gage should not be graduated in pounds above a perfect vacuum, and its units used for whatever pressures are measured. U-tube scales, no matter what their liquid, could be graduated to read directly in pounds per square inch quite as conveniently as the present system of inches. It may be contended that for drafts and small pressure differences the pound is too large a unit. This could be remedied by using one one-hundredth of a pound as the unit for small pressures. There would then be 3.6 of these units to an inch of water.—"Power."

PRIMING IN WATER-TUBE BOILERS.

As every one of the many different types of boiler has its peculiarity and particular defect, priming may result in any boiler from a diversity of causes, which may apply to one type of boiler, but is inapplicable to that of another type, and so far as I am aware no one has as yet successfully overcome its evil effects or provided an efficient remedy, although numerous arrangements of collecting pipes and priming traps, separators, etc., have been frequently used, but their novelty and expense in first cost, repair and upkeep generally exceeds their usefulness.

Priming consists of the lifting of the water in the boiler into the steam pipe, and it is analogous to boiling over. It is caused in many ways, viz: insufficiency of boiler power, defective design of boiler, or being too full of tube surface, dirty water, irregular firing, etc. When the steam is actually surcharged with water or becomes a watery vapor prompt attention should be given to cylinder and casing drain cocks, otherwise there is danger of splitting the cylinders or knocking the covers off or bottoms out. Priming in any quantity is a loss of economy, although it may not be perceptible or dangerous.

In water-tube boilers ashes accumulate among the tubes, and cannot, unless special doors are fitted, be cleared when under way, so that whenever the engines are eased or stopped the spaces between the tubes should be swept and cleared as far as possible, and, as opportunity offers, the surfaces should be cleaned. Soot is a very bad conductor of heat, and consequently any of it on the heating surfaces considerably reduces the evaporative power. The water should be as pure as possible, and with the evaporators now generally fitted in vessels this is not a very difficult matter. Salt and impure waters are a frequent cause of priming. The use of either can be guarded against except in extreme cases, and even then if they are admitted in minute quantities, as make-up or extra feed, the danger of priming is reduced, but it cannot in some boilers be entirely overcome. Priming generally comes from irregular working more than from the nature of the feed water. In boilers with a sediment, or mud box, this box should be blown out frequently, and in other boilers the judicious use of the extra feed valve reduces the chances of priming and scale forming considerably. Ordinary or common steam frequently contains as much as 30 per cent. by weight of priming water, and this may be taken as the safety limit, any higher percentage being dangerous.

The principal known causes of priming are:

1. Bad circulation of water in the boiler. The only remedy for this is to promote a good circulation by proper design of the boiler, or by artificial means such as the hydrokineter. The circulation should be continuous, and a downward current should be provided for, so that on all occasions the direction should be maintained at all points. This is found in most of the water-tube boilers of the "express," or small-tube type, and also in the Belleville boiler. In flame-tube boilers it is difficult to arrange for a distinct circulation, generally it is made by dividing the combustion chamber into several parts, so that each furnace has its own particular combustion chamber. Between these combustion chambers is a water space leading up from the bottom of the boiler. The subdivison of large combustion chambers by brickwork is useless in this connection, and is of doubtful utility at any time. Bad circulation is not so marked in flame-tube boilers as in water-tube boilers; this is due to the much greater quantity of water in the boiler and larger surface area from which the same quantity of steam is generated.

2. Contracted steam space. This space is the volume of the boiler internally above the working water level, and can only be increased by increasing the size of this part of the boiler. The effect is, however, minimized by using a higher boiler pressure than engine pressure by wire-

drawing the steam; a reducing valve is now generally fitted with nearly all water-tube boilers.

3. Contracted water-surface area. The quantity of steam which can escape from a give surface, without carrying off an undue amount of priming water, must be some tangible amount and there appears to be no difficulty in its determination by experiment. This quantity varies with the condition of the water used; it is greatest with clean fresh water, and probably least with a mixture of fresh and salt water. The various types of screen and separator plates fitted in the steam space of some water-tube boilers are really an artificial method of increasing the otherwise restricted water-surface areas. The surface area in the majority of water-tube boilers might be increased with advantage, and to ensure greater economy; the separator plates might also be improved, and with the exception of the upper part, which takes the place of an internal steam pipe, they are of doubtful utility.

4. Irregular working. If an automatic feed arrangement is fitted, it may possibly stop excessive priming from an irregular feed supply, and it is a noticeable feature of present practice that automatic feeding arrangements have secured almost general adoption. With the large-flame tube boilers it is not necessary, the percentage of the feed water entering the boiler being very small compared with the water contained in the boiler, consequently it does not materially affect the water level and steam pressure. Bad stoking comes under the head of irregular working and it is an important factor when there is any original tendency to prime from other causes, especially from dirty or impure water.

5. Sudden increase of the revolutions of the engine. Priming from this cause is principally dependent on the watchfulness and care of the engine-room staff. The output of the boilers can be limited by partially closing the stop valves on the boilers, and also on any bulkheads in connection with the main steam pipe. Engine builders generally allow a large margin for emergencies and consequently the steam valve is larger, with a correspondingly excessive lift, than is ultimately found necessary. By opening these valves to a known limited extent only, priming can be greatly reduced. Reducing valves act in a similar way, and have also the advantage of being automatic. A calculation giving the amount of lift may be based on the following data:

Coal burnt per I.H.P. per hour $= K = 2$ pounds for naval (about), or $1\frac{1}{2}$ pounds for mercantile marine.

Water evaporated per pound of coal $= 8$ pounds.

Volume of 1 pound of steam at working pressure $= V$. (See tables of absolute pressure, temperature, volume and density of steam.)

Let $c =$ circumference of valve orifice, in inches.

$x =$ lift of valve in inches.

Then $c \times x =$ area through valve orifice.

The velocity of steam through valve is about 8,000 feet per minute.

Then volume of steam which passes through valve per minute $=$

$$\frac{8{,}000 \times c \times x}{144}$$

and the weight $=$

$$\frac{8{,}000 \times c \times x}{144 \times V}$$

and this must be equal to the weight of water evaporated in the same time $=$

$$\frac{8 \times K \times \text{I.H.P.}}{60}$$

therefore

$$x = \frac{8 \times K \times \text{I.H.P.}}{60} \times \frac{144 \times V}{8{,}000 \times c}$$

$$= \frac{3 \times K \times V}{1{,}250 \times c} \times \text{I.H.P. in inches.}$$

6. The condition of the heating surfaces. To obtain the best evaporative results, it is necessary to keep the heating surfaces as clean as possible. In boilers with bad circulation extreme cleanliness may cause priming locally, and it has not been unusual to find that although priming took place when the boilers were new (*i. e.*, perfectly clean), they did not prime subsequently, after a slight scale had been formed. From this experience it might be inferred that a slight scale would check excessive priming, and this is so in flame-tube boilers, but it is impossible to obtain an equally great evaporation as with clean surfaces from the same boiler, and if we are to obtain the best results, good circulation must be combined with clean surfaces. The sea-going engineer cannot make a mistake in keeping his boilers as clean as it is possible. Certainly without clean heating surfaces he cannot expect to realize the highest economy. It is possible that a near approach to good results may be obtained with a slight scale on the furnaces or other heating surfaces, but at any time this scale may cause a come-down or blister on the furnace crown or combustion chamber; priming is preferable to this. With water-tube boilers dirty heating surfaces on the water side will cause priming at a very low evaporative power; this is due to the scale becoming detached, not to scale adhering to the heating surface. In either case it is a loss of evaporative power.

7. Salt or impure water. Without doubt this is the greatest cause of priming in all boilers, and although for other reasons it may be needful to admit salt water in small quantities to the boiler, it should be very carefully watched when high evaporative results are expected. The presence of clean salt water is not shown until a period of working is reached, which varies with each type of boiler; this variation is nearly in proportion to the steam space provided, combined with the surface area of the water at working height. Priming would be also likely to occur when the evaporation per square foot of heating surface reaches 3½ pounds per hour. By very gradual increase of the speed, taking care that the steam pressure is not decreased at the same time, the priming may perhaps (and there is some experience to prove this) be kept within the limits of safety and the evaporation increased to 7 or 8 pounds, or slightly under that obtained with pure distilled water. Priming does not usually occur when the steam pressure is rising, and with very careful management in increasing the speed, it may be reduced to a certain point which is very accurately defined in certain types of boiler, the various types differing much less than might be expected from their various designs. In regard to naval vessels, it has been stated that certain experiments showed about three-fourths of the speed of a 27-knot destroyer can be obtained with salt water. This represents a loss of nearly 7 knots, and more than half the power, and it is probably a high estimate. There is no necessity to use salt water on actual service to such an extent that the tubes become wholly choked with scale; if this happens every tube so choked is in imminent danger of bursting. In 1897 a first-class torpedo boat was steamed from the builders' yard on the Thames at London, England, to South America, salt water being used for the last two weeks of the voyage, and the boilers were not injured in any way, an official trial being made after the arrival of the boat. The speed at which this voyage was made was certainly a very low one, prob-

ably ten or twelve knots, and had it been necessary, at any time after salt water had been admitted, to increase the speed to, say twenty knots, it could not have been done, although twenty-four knots was obtainable with clean fresh water.

The writer was connected with a case a few years ago which abundantly proved the evil effects of using salt water in water-tube boilers either designedly or otherwise. Two destroyers had been built to the order of the Japanese Government on the Thames, England; delivery to be made in Yokohoma, Japan. Each of the vessels was capable of a speed of about twenty-five knots, and was fitted with four-cylinder triple-expansion balanced reciprocating engines and boilers of the Yarrow type. With a picked crew and experienced engineers the destroyers left for the far East, but in crossing the Indian Ocean their evaporators gave out, added to which the main condenser in one of the vessels became leaky and continued blowing down of the boilers had to be resorted to, with the inevitable result of salting up the tubes. Repairs were attempted several times at sea, and also at the coaling stations, but proved ineffectual; when eventually the vessels arrived at Yokohama, after a very much delayed voyage, the boilers were in a deplorable condition; many of the tubes had been plugged, some had been cut out altogether, and others were practically solid with salt and scale. Extensive repairs and renewal of tubes had to be made at considerable expense before the Japanase Government would accept the destroyers. To retube a Yarrow-type boiler takes considerable time, and is costly, as from the construction of the boiler all and every joint and pipe connection on the boiler has to be broken, and the boiler itself lifted a sufficient height to enable the tubes to be passed out through the bottom drums.

In the final analyses, therefore, it is found that if a water-tube boiler of any type can always be steamed, with perfectly clean fresh water a maximum of power may be obtained at any time, but if at any moment salt or dirty water gets into the boiler the speed and power are considerably diminished, and probably the greatest power is little more than one-half that obtainable with fresh clean water.

Such conditions are unsuitable for war vessels, and are certainly not good enough for war service. Too much is expected from water-tube boilers generally, and there seems to be a tendency to rush to extremes; because it has been found that engineers have been able to cope with the difficulty of maintaining their boilers in a high state of efficiency by enormous personal effort during times of peace, it does not follow that in war time, when the opportunities for cleaning and refilling boilers will be very infrequent, that this state of efficiency can be maintained, and it is possible that a vessel capable of a speed of 27 knots, may be reduced to less than 20, because a percentage of salt water, or some other priming mixture, has been admitted into the boilers. Pure fresh water cannot always be maintained on long cruises, and under the varying conditions of warfare this must be exceptional. The present safe limit appears to be "natural draft" (equal to ½ inch air pressure), this will work out at about 50 horsepower per ton weight of boiler. Under these conditions the contract power may be relied upon, and perhaps a greater power may be developed for short periods in emergencies. Another reason for the adoption of limit is that the greater forcing of the fires, the shorter the period at which the power can be maintained. If the speed and power are to be maintained for longer than eight hours, the rate of combustion per hour should not exceed 30 pounds of coal per square foot of grate surface. Within this limit there is still a gain of about 50 per cent. on the older types of boiler.

Not the least of an engineer's troubles is the difficulty of finding a really automatically working evaporator which will not prime when left to its "automatic" self for a few minutes.

A perfectly reliable non-priming evaporator may exist, but personally I

have not found one when working in conjunction with water-tube boilers. A slight inattention on the part of the engineer, and priming commences, impregnating the feed water with salt water, and generally such a mixture causes intermittent and excessive priming of the main boilers. When the water in the boiler becomes slightly acid it is usual to correct this acidity, and consequent liability to corrosion, by injecting a small quantity of soda or other alkali. With the old type of boilers a large quantity of soda was used from time to time, in proper proportions to the feed admitted, this amount as a rule had no priming effect. In most water-tube boilers, however, it is a dangerous priming mixture, and the engineer has to decide on the lesser of two evils, priming, and consequent reduction of power or corrosion; generally priming is selected, and some soda is injected with the feed water, a proportion of rather under half a pound of soda to each ton of water contained in the boiler being used. After a day or two's steaming the boilers have to be emptied and refilled, and perhaps for a few days no sign of acidity is shown, but eventually the soda process has to be repeated; further, in some boilers no sediment chamber and fittings have, as a rule, been fitted, and consequently it is necessary to empty the boiler into the bilge to the detriment of the framing and plates of the hull of the ship; this entails painting these parts throughout frequently. By adopting the sediment-chamber system much of the priming matter could be blown out while under steam, thus assisting in the maintenance of the efficiency obtained on the original trials run under the best conditions of clean fresh water in clean boilers.

"Priming" and "forcing" go hand in hand, except that under certain very favorable conditions "forcing" can be used without priming occurring. Under a forced draft of 3 or 4 inches of air pressure the extreme power may be obtained for perhaps four hours, if priming does not occur. About 90 to 100 I.H.P. per ton weight of boiler is a possibility with 3½ to 4 inches of air pressure. Perhaps in a destroyer this may be admissible, such extreme speeds as 30 knots may not be required for more than 4 hours, but when a similar air pressure is used to obtain the bare 20 knots of a cruiser four hours is far too short a period. To counteract this tendency a limit should be placed, not on the air pressure but on the I.H.P. to be obtained per square foot of grate surface. With half an inch of air pressure it is possible to burn 30 pounds of coal per square foot of grate, and this will produce about 15 I.H.P. In a destroyer it is necessary to burn 70 pounds per square foot of grate when steaming the contract speed.

Priming plays an important part in the lack of economy obtained with some boilers, and probably accounts for part of the comparatively high rate of coal consumption; the method adopted of fitting a reducing valve at each engine only, cannot be efficient in reducing priming, as when a reducing valve is fitted on each boiler. In ordinary practice when priming occurs the speed of the engine is reduced until a safe working limit is reached; but if the speed must be maintained, the practical man partially closes the boiler stop valves, thus regulating the output of each boiler, and within certain limits previously noticed this checks some of the priming. When priming commences in one boiler more work is thrown on the remainder, and consequently it may induce priming in some of them.

One problem still to be solved in connection with priming is that under certain conditions one boiler may increase its output almost indefinitely even when independent reducing valves are fitted; reducing the pressure is insufficient, unless the output is also limited. "False water" is a term applied to the apparent increase of the volume of the water in a boiler when priming takes place. It may be imperceptible; but it often causes an apparent rising of the water level to the extent of several inches.

In considering the final efficiency of the coal burned, water-tube boilers are at somewhat of a disadvantage, owing in the first place to the infiltration of cold air through the outer casing, thus diluting and cooling the heated products of combustion. The extent of this will depend much upon

the care with which the casing is fitted, and with reasonable care it may be reduced to practically a negligible amount. There is also a further disadvantage that under equal conditions the heating surface does not seem to be quite as efficient per square foot as in the fire-tube boilers. In the former the current of hot gases sweeps around the outside of the tubes, and consequently the leeward side of each tube is more or less shielded from the direct inpingement of the heated gases, and therefore cannot be expected to be as efficient as the front side. With flame tubes the interior of the tube is more uniformly bathed with the hot gas, and its average efficiency is somewhat higher. So far as the final result is affected, this is a difference which can be overcome by providing a relatively greater heating surface in water-tube than in fire-tube boilers, other conditions being the same. The amount of this excess is not definitely shown by any available data, but indications point to an amount varying from 10 to 20 per cent.

The efficiency per pound weight of boiler. The water-tube boiler in this regard scores perhaps its greatest advantage. This arises from two causes: (1) A less weight per pound of steam delivered to the engine; (2) the possibility of higher pressures and hence, if chosen, of a higher mean effective pressure in the engine with a reasonable number of expansions, and hence a smaller and lighter engine for a given power developed. The boiler itself is lighter, because the outer shell of fire-tube boilers, solid steel and from 1¼ to 1¾ inches thick, is replaced by a comparatively thin, single or double casing of sheet iron, thus saving greatly in weight. Again, the tubes on the whole are smaller in diameter than with fire tubes, and may therefore be made slightly thinner. The gain here is slight, because considerations of general solidity of work, provisions against corrosion, etc., prevent the decrease in thickness beyond a certain point. The absence of bracing in water-tube boilers also conduces to lightness. In many such boilers fire brick is freely used in the furnaces for lining and to prevent the lower ends of the tubes from too close contact with the fire. This item is usually in excess of any corresponding item in the fire-tube boilers and hence constitutes a slight offset to the general excess of weight in the latter.

Passing from structural features to the amount of contained water, we find for the water-tube boiler its most important gain in point of weight. In fire-tube boilers the contained water is usually about 40 per cent. of the total, while with the water-tube it is not usually more than 12 to 15 per cent. On the whole, we find that boilers of the former type weigh from 35 to 60 pounds per square foot of heating surface, while those of the latter weigh from 12 to 20. Or, including the engine in the comparison, we find that from 20 to 30 I.H.P. may be developed per ton of fire-tube boilers, and from 40 to 70 per ton of water-tube boilers. The range of these figures is, of course, large, as we should expect from the varied conditions under which the trials are made, and the multitudinous varieties of boilers of the latter type. For equal power developed we may expect water-tube boilers to have from somewhat more than one-half to perhaps as little as one-third the weight of fire-tube boilers under the same conditions. This gain is, of course, independent of the possibility of further gain, from a decrease in the size and weight of engine through a higher range of pressures throughout. In the operation, derangement, durability and repair of water-tube boilers, it is difficult to make satisfactory comparisons under these heads, because these boilers exist in so many varied forms, and the peculiarities of one are entirely distinct from those of another. On the whole, however, it is found that under operation an advantage for the water-tube boiler, in the rapidity with which steam may be raised; half an hour, if need be, being sufficient in this case, whereas 3 or 4 hours or longer (depending on the size and condition of the boiler) would be required for the other type. It is not unusual to allow 12 hours to raise

steam in the Scotch marine boiler from the time of lighting fires, to allow for the gradual and uniform heating and expansion of all parts. Again, the water-tube boiler in general stands hard forcing without as great likelihood of leaky tubes, undue internal strains, etc., as exists with the fire-tube boiler under like conditions. On the other hand, the small amount of water contained in these boilers makes it necessary to pay the strictest attention to regularity of feed, and renders the liability to serious results, from a temporary stoppage of the feed, somewhat greater. This point is one that is considered of great importance, and much thought has been given to insuring a regular supply of feed. In case the feed supply is irregular, this may lead on the one hand, with a scarcity of water, to the burning out of tubes, or, on the other, to a flooding of the tubes and upper chamber, and to a carrying of the water with the steam. In addition to great moisture of steam some water-tube boilers have given trouble with great unsteadiness of the water level, and with more or less violent priming. In some cases this has led to the necessity of fitting special separators between the boiler and the engine, in addition to the separating chamber always a distinct feature of this type of boiler. These difficulties are undoubtedly due to peculiarities of design and proportion, rather than inherent in the type itself, for there are numerous instances where such boilers have performed continuously for considerable periods under high forcing with steady water level, delivering dry, and in some cases superheated steam.

As another disadvantage for deep-water service, we may note the restriction to fresh water for feed. The interior of the relatively small water tubes cannot be so readily cleaned of scale as the interior of fire-tube boilers, and, in spite of various means intended to localize the deposit to the bottom drums, the tubes will soon become coated if salt water is used as feed. A deposit of scale in water-tube is, moreover, still more undesirable than in the fire-tube boilers, from a likelihood of a greater disturbance to the circulation and a greater liability to overheating. This necessity of a fresh-water feed is, in large measure, met by the growing belief that no boiler should, under ordinary regimen, be fed with salt water, and that all deficit in fresh feed should be made up by evaporators, distillers, etc. As to the liability to serious results from oil in the feed water, there is perhaps little choice between the two types of boilers. In the case of any boiler fed with oily feed water there will be a liability to collapsed furnaces, burnt-out tubes, etc., and as far as it is practicable the oil should be removed by filtration before the feed enters the boiler.

As to durability, deterioration, etc., so much depends upon the especial type of boiler, and the conditions of use, and further, on the comparatively short period for which most of the water-tube boilers have been known, that here again a satisfactory comparison is difficult. All evidence, however, seems to point toward the same life for the tubes in one form, or in the other, other conditions being considered equally fair. In the matter of the case of cost of repair, it is found for same reasons as above, difficult in making any comprehensive comparison, and it appears that the difference under this head will depend more upon the design of the boiler in question than on any inherent difference in the two types. The care of a steam boiler of any type, and its proper and efficient management, demands much knowledge and experience of the causes and methods of prevention and of remedy of injury, on the part of those to whom it is entrusted; such general care includes the preservation of the boiler against decay, or loss of efficiency, the retention of all its parts in good repair, and the maintenance in good order of all its accessories and connections. This calls for the highest skill and constant watchfulness, not only when the boiler is in operation, but also when the boiler is not in use, as it is principally then that corrosion in some of its various forms may appear.

JOHN CREEN in "The American Marine Engineer."

LEAD-LINED PIPING ON SHIPBOARD

Description of its Manufacture.—In repairing pipe lines brass, iron and copper piping may be lined, but for new work lap-welded steel casing (being much cheaper than seamless-drawn tubing and equally as satisfactory) is used. The inner surface of this casing is first made entirely smooth by a revolving chain mop on a flexible spindle attached to a pneumatic drilling machine. In the case of old piping this is first cleaned and the pits closed by solder, wiped off smooth, and tested to a pressure of 300 pounds.

All joints are made with flanges. When repairing a pipe having several different bends it is sometimes necessary to cut the pipe at the bend and to fit flanges. The flanges are counter-bored to about ¾ inch from the face of the pipe, depending on the size of the flange, and about 1/10 inch deep, or the thickness of the lead which is to be flanged over, *care being taken that the edge of the casing over which the lead is bent is carefully rounded with a file.*

The lead tubing used is about ¼ inch smaller than the pipe to be lined Where bends are encountered it is necessary to pull the lining through. This is done by means of a rope led through the lead tube and then through a plug and secured by a knot. The lead tubing is then filled with dry sand, covered with tallow and, while the outer tube is tapped with a wooden mallet, pulled through by means of a machine operated by air or water pressure at from 15 to 100 pounds pressure, according to the nature of the bend. Bad bends are warmed slightly by means of a gasoline torch.

The plug, sand and rope are then removed and the lead examined by inserting a small electric portable. If no defects appear the tubing is tapped over into the recess of the flange at one end of the pipe only. The other end of the lead pipe is sawed off about 6 inches longer than the pipe that is to be lined and a plug is fitted.

The air cock shown is opened and the pressure turned on slowly. When the air is all out the plug is closed and the pressure very gradually run up to 300 pounds. The lead pipe becomes shorter under this pressure which is then removed and the lead fitted into the recess of the flange at the other end of the pipe. The piece of piping is then blank-flanged and treated to a final pressure of 600 pounds per square inch.

Weight and Cost.—In order to reduce weight the lining should be no thicker than absolutely necessary. For sizes given in the following table the weight and thickness there given have been found very satisfactory:

Outside diameter of pipe.	Weight of pipe.	Thickness.
1½ inch	2	.088
2	3	.099
3	4	.088
4	5¾	.095

The weight of such a system is about 5 per cent. less than that of a similar installation of copper piping. The cost is about 50 per cent. less.

Further Points of Interest Thus Far Determined.—1. The lead piping should be shipped from the works of the manufacturer very carefully packed on wooden mandrels to prevent crushing and distortion.

2. In the case of galvanized-iron, wrought-iron or steel tubing, the elbows, tees and other fittings are not lead lined. It has been found necessary to lead line fittings under no circumstances.

3. Old pipe has been lined with lead where the radius of the bend was twice the diameter, but it is better practice to bend the tubing to be lined to about 4 or 5 times the diameter.

4. In relining old copper pipe it is much cheaper to reline the old pipe even though it is necessary to cut out the brazed flanges and substitute flanged fittings.

5. The process does not work well on pipe above 5½ inches on account of the crushing of the lead tubing when drawing it into the steel casing.

6. Lead-lined piping is not adapted to suction pipes. The suction pulls the lead away from the walls of the steel casing.

7. Lead-lined piping is not adapted for use with very hot water or steam service, such as bottom-blow piping for evaporators, on account of the large expansion in those cases.

8. Thickness of tubing thinner than that given in the above table should not be used. Considerable difficulty has been experienced on recent torpedo-boat destroyers in the attempt to save weight in this manner.

9. Special care should be used to give no crown to the lead where expanded over the recess of the flange. Unless a perfectly flush surface is thus obtained it has been found that in setting up the joint the pressure comes on the lead, causing it to separate from the flange recess or to crack where bent over the steel casing.

10. Several cases of severe lead poisoning have occurred at Mare Island due to the milky oxide of lead on the outside of the tubing getting into the pores of the skin of the workmen, and all the usual precautions should be taken to guard against this form of poisoning.—"U. S. Naval Institute."

THE FOLLOWING were found among rejected manuscript of the "Confidential Bulletin," a publication issued by the Bureau of Steam Engineering to officers performing engineering duty.

Calorific Conservation Contest.

During the last fiscal year in each pound of fuel issued to vessels of the U. S. Navy there were 14,370 B. T. U.s. Of these, 3,613 were destroyed in doing useful work, but 10,757 were allowed to escape and are still at large.

Careless custodians!

The total of this loss of energy in the vessels of the Navy is sufficient

a. To run Niagara Falls backwards.

b. If applied to propelling machinery in the Dewey Drydock, to give the dock a speed of 27 knots, thus permitting its use as a scout cruiser or as a commerce destroyer.

c. Or to do other similarly useful work.

To encourage the conservation of this hitherto wasted energy, the "Bulletin" will award monthly prizes as follows:

To the engineer officer who sends in, properly counted and packed, the largest number of heat units captured while attempting to avoid useful work, a pair of dungaree trousers (almost new).

To each of the five next most successful contestants, a year's subscription to the "Bulletin."

We want Conservation, not Conversation!

Each ten thousand heat units must be accompanied by a coupon which will be found on the last page of the "Bulletin."

Shipments should be prepaid and addressed to "The B. T. U. Man, Confidential Bulletin, Bureau of Steam Engineering, Navy Department, Washington, D. C."

Installation of Incinerators.

Experiments at the morgue have demonstrated that the human corpse contains B. T. U.s varying from 427,200 in a person weighing 120 pounds to 1,236,400 in an obese person of 270 pounds.

To enable the patriotic corpse to devote this potential energy to the service of his country, incinerators are to be installed on all large ships.

Remains of the decedent will be forwarded to his next of kin in a penalty envelope.

RESULTS OBTAINED BY THE FRENCH NAVY IN SUBMARINE NAVIGATION.

Translated from the [Chilian] "Rivista de Marina," March, 1911.

In France submarine navigation, created by naval engineers, improved and put into practice by naval officers and crews, has been jealously confined, until the last few years, to naval departments. There is, therefore, little known about it generally, and its efforts and successes, which are largely in excess of those obtained by other foreign navies, have not attracted much attention, nor is there any special publication devoted to the subject. But now that the Navy has had recourse to the efficient help of national industry,* it is right that the results obtained, which have now become accomplished facts, should be made known.

It is to the unwearied efforts of their commanders, for the last twenty years, that is due that perfection of navigability to which the French submarines have attained, both on the surface and in deep water, so that a doctrine now exists, accepted by all without discussion, and progress is limited to seeking improvement in speed and armaments, as well as to the study of the tactical utility.

In order to form a sound judgment of the value of submarines, it suffices to glance through the reports on the numerous exercises carried out in France, and to examine the results which such maneuvers would obtain in war. For more than two years past the French flotillas have been completely organized; the repairs, inspections, periodical overhauling in the docks, take place normally, and the program of exercises, prepared with much forethought, follow their regular course.

The anxiety to ensure the safety of the matériel has been overcome. The endurance of the personnel is alone taken into account.

During one week out of every four there are carried out in every flotilla exercises in attack, and in firing against an armor-plated vessel, which serves as target. The discharge "à cone de choc," which is an indisputable criterion of a good attack, is regularly practiced. The target moves in the sectors occupied by the submarines, with varying velocity, in unexpected directions, and on occasion has the right of acting in the manner best calculated to avoid any periscope that has been sighted.

In spite of these severe conditions, in the competition of honor of 1909, the torpedo fire gave the following results for the Cherbourg flotilla—eight submarines took part, of which six were "Laubeuf" submersibles.

Four "Laubeuf" submersibles headed the list. Not one periscope was sighted, in spite of extreme vigilance.

In 1910, the same results. The *Ventose* and the *Germinal* were the most brilliant.

In the defensive operations, of short duration, submarines of every type and tonnage may be employed. But the same is not the case when the object is to attack an enemy who is far from his own bases of supply. The following experiments have demonstrated with complete precision that in the latter case only vessels with a tonnage of three to four hundred tons are of any military importance.

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* Alluding to the Creusot Establishment.

MANEUVERS OF MAY AND JUNE, 1909.

The first important tactical exercises which took place in France date from May, 1909. At that time the question of the type of vessel which should be selected was no longer under discussion, because, after long experience of the type *Narvel* and its derivatives, the Navy had definitely adopted the submersible of the "Laubeuf" type.

The first vessels of the *Pluviose* type had then completed their trials, and the subsequent ones had profited by all the improvements of details which had been recognized as necessary in their various experiments.

The vessels which took part in the first exercises were the *Pluviose* and the *Ventose*, the *Esmeralda* and the *Opale*. The two first belong to the latest "Laubeuf" type of 400 tons displacement, 27 per cent. of buoyancy, resembling floating ships with their exterior ballast tanks, naphtha boilers and their steam machinery.

The *Esmeralda* and the *Opale* belong to the "Maugras" type; they are derived from the *Farfadet*, with a displacement of 300 tons, are cigar shaped, with a very low freeboard with their 7.5 per cent. of buoyancy, interior ballast tanks, and Diesel petrol motor. These vessels, which date from 1903, have undergone exhaustive trials. The exercises ordered were not originally intended as comparative trials between these two very different types, but only to ascertain whether under war conditions, it was possible to obtain from these vessels all the requirements which one had a right to expect from their tonnage and armaments.

The conditions for diving were more or less similar; those for movement on the surface, with advantage to the submersibles.

The maneuvers included three successive phases:

1. Closing the Straits of Dover in order to prevent the passage of a hostile fleet coming from the North Sea.

2. Defence of Lorient and of Quiberon, as well as of places in the vicinity suitable for disembarking troops, against a fleet which had blockaded those ports.

3. Blockade of the port of Cherbourg, the enemy's base, where a naval force had taken shelter, and whose departure was to be prevented.

Thus exercises were carried out in the three great objectives of a flotilla of offensive submarines, with a large radius of action, viz: control of a narrow sea by closely watching the entrances; defence, against a hostile menace, of a national port situated far from the base of the flotilla; and lastly, close blockade of a hostile fleet within its own port.

First Phase.—The northern fleet represented the enemy, who came from the North Sea. The coast of France was friendly to the submarines, and the semaphores and the torpedo boats of that region tried to give information to the submarines as to the whereabouts of the enemy. In spite of a tolerably thick fog, which was rather favorable to the fleet, two ironclads were attacked with torpedoes by the *Pluviose*.

The *Ventose* attacked an English cruiser by mistake. At the moment of emergence she was sighted by the destroyers of the fleet; being compelled to dive again, and so evade pursuit, she allowed the fleet to pass without being able to attack it.

Escorted by the *Valmy*, the four submarines made at once for Cherbourg. Off Barfleur, the officer commanding the operations signaled "Alerte," and the submarines dived simultaneously. The two submersibles disappeared in four minutes, the *Esmeralda* in four and a-half minutes. Having weighed anchor at 3 A. M., these little vessels, after having been at sea without interruption, returned to Cherbourg on the night of the 5th-6th, in perfect condition, the crews active and fresh, in spite of a stiff breeze from the East. The matériel showed no sign of wear and tear.

Second Phase.—The four submarines started from Cherbourg at 10 A. M., and reached Lorient in 34 hours, delayed slightly by the heating of

the machinery of the *Opale*. The submarines remained until the 6th July, during the day at Lorient and its proximity, attacking the Northern Fleet each time that the latter attempted to approach. Surprised by the fog, the fleet was compelled to anchor on May 21st, and protected itself with a powerful boom and screens of numerous destroyers and steam launches. Nevertheless, the fleet was attacked at its anchorage by the submarines, which, evading the vigilance of the enemy, and, passing under the boom, succeeded in attacking the units with torpedoes.

The *Pluviose*, in diving, struck a boom, against which she injured her periscope, passed the obstacle rapidly, and finished the attack with her second periscope. On her return, she repaired her periscope, and resumed the exercises on the following day.

From the 16th May to 4th June the submarines, which remained at sea two days out of every three, succeeded in making some forty attacks. In the opinion of all the commanders of the Fleet, the situation of the latter was untenable, and yet the Fleet had only to deal with three submarines, because the *Opale* had been rendered useless from the outset, owing to damage to her machinery.

Third Phase.—The submarines *Pluviose*, *Ventose* and *Esmeralda* left Lorient for Cherbourg, which port they blockaded at once. During three days and three nights they dived during the day, so as not to be seen by the scouts and the semaphores; they recharged their accumulators at night on the surface to avoid the destroyers which pursued them day and night.

The hostile fleet tried several times to escape. At every attempt they were attacked and torpedoed, without any destroyer having been able to surprise a single submarine. At the end of the third day of blockade (the limit fixed, in order not to exhaust the personnel) the three vessels made for Dunkirk without touching land, and returned to Cherbourg without stopping anywhere.

They remained at sea uninterruptedly for six days; before and after the blockade they made two voyages of 400 miles, and 48 hours of immersion during the three days of blockade. Such a demonstration needs no commentaries. As was to be expected, the submersibles have shown themselves clearly superior to the submarines as regards seaworthiness, and, above all, as regards the accommodation for the personnel. The personnel of the *Esmeralda* had reached the very limits of endurance, while the submersibles could have still proceeded on a longer cruise.

Exercises and Preparation.—Every time that a squadron is in the neighborhood of military ports, the opportunity is taken to prepare a series of exercises for the flotilla. The defenders occupy the nearest posts, and the more distant sectors are reserved for the "400 tons." At Toulon, Rochefort, Biserta, Calais and Cherbourg there was never occasion to record the slightest damage to the matériel, nor any serious incident laid to the account of these same vessels. On the other hand, the matériel, submitted to severe trials of endurance, has stood them well. At Cherbourg, every vessel, on receiving its armament, has to accomplish, on the surface, the tour of Cherbourg, Brest, Dunkirk, Cherbourg—in other words, 700 miles. On its return, it has to undergo an immersion of long duration, fire its war torpedoes, and recharge its batteries by its own resources.

In pursuance of a general program for four weeks, the submarines leave their port of origin in turn, and without convoy, for the pilotage of the neighboring coasts, anchoring on the open coast, or in a river, entering and leaving and mooring at the wharves of commercial ports, like ordinary torpedo boats.

We may mention here the cruise accomplished by the *Papin* in December, 1909. The *Papin*, a submersible of the *Pluviose* type, fitted out at Rochefort, was sent to Cherbourg to place herself in communication with the numerous "Pluvioses" already equipped. She stayed there eight days, returned to Rochefort, from whence she proceeded to Oran, then to Al-

giers and Biserta; in other words, a total, in less than three weeks, of 2,300 miles, without fatigue and without incident. The voyage from Rochefort to Oran was made without a break and in bad weather.

GREAT NAVAL MANEUVERS IN 1910.

The different schemes proposed had no further object in view than squadron maneuvers; nevertheless, no one could fail to be surprised by the unexpected part played by the three large submersibles of Biserta—namely, *Papin* (*Pluviose* type), *Circe* and *Calypso* (*Pluviose* type with Diesel motors), attached to the Red side.

They steered from Biserta to Ajaccio, with orders to prevent the Blue Fleet from blocking the latter port. Every time that the Blue Squadron attempted to approach the port, in order to bombard it, one at least of the submarines attacked it, firing torpedoes against various units. In the face of this constant menace on the part of the submarines, the squadron was compelled to withdraw to such a distance from the port that the blockade became almost a farce.

The Blue Squadron became aware of the presence of the Red to the south of Sardinia, while traversing at 18 knots the Strait of Bonifaccio for the purpose of fighting the enemy; the squadron was attacked by two of the submersibles, which had left Ajaccio, passing below the screen formed by the Blue destroyers.

All the French and foreign journals united in praise of the performance of these submersibles.

The following extract from the *Novoye Vremya*, of 4/17th August last, over the signature of "Brutus," is interesting:

"A squadron of six armored vessels passed through the Strait of Bonifaccio in close column, and keeping a distance of 400 meters from mast to mast. This formation was arranged with the intention of giving the squadron the possibility of escaping more quickly from the danger of an encounter with submarines than it would otherwise have had. At the same time the squadron was surrounded by two similar columns, each consisting of six torpedo boats, for the protection of the ironclads against the lateral attack of the submarines. Moreover, these three columns advanced at the extraordinary speed of 17 knots an hour, while in front of the squadron a few swift destroyers ran a zig-zag in order to repel or sink whatever might impede the way.

"It is evident that, as regards peril, the submarines were under conditions similar to those in war; the slightest error on the part of the commanding officer would inevitably have entailed the loss of the submarine. Indeed it seems strange that such experiments should take place in time of peace. But all the world was persuaded that, under these conditions, the presence of the submarine was useless, since not one would ever venture to emerge from the water close to a fleet. In accordance with this conviction, the partisans of the armored fleet reckoned that none of their opponents would have the temerity to accept this challenge. But they were mistaken.

"Two submersibles, which guarded the coast of the Strait of Bonifaccio, dashed forth as for attack, and one of them, the *Papin*, rose to the surface between the columns of the squadron, close to the third ship from the rear, *La Verite*, aiming at her its torpedo apparatus. In consequence of this maneuver the *Papin* would have to be reckoned out of action, lost, but probably in a real battle the submarine would have had sufficient time to torpedo also the last ship of the column, the *Justice*, which followed the *Verité*.

"'The appearance of the submarine struck me speechless,' said an Admiral, hitherto a disbeliever in the submarine flotilla. 'I never could have

imagined anything like it, it is evident that armor-clads are no longer sole masters of the sea.'"

In short, during these maneuvers, which lasted one month and a half, these three little vessels gave evidence of great seagoing qualities, and of noteworthy endurance, both of matériel and of personnel.

CONCLUSIONS.

The three large submersibles of the "Laubeuf" type combine the following qualities: they are very seaworthy; they rise perfectly to the surface, owing to their great buoyancy; the accommodation for the crew is such that, in spite of necessary imperfections, the men prefer these vessels to the torpedo boats. Good conditions for rest while at anchor are assured, by the extensive bridge and the four windsails, which produce considerable natural ventilation. This condition is essential, because it is necessary that wherever they may be, after long voyages, and all the hardships of immersion, the crew should have the means of repairing their strength under the best possible conditions of hygiene and comfort.

The *Pluviôse* vessels, not being encumbered by interior ballast tanks, fulfill this condition completely. Another great advantage obtained by the Diesel petrol motors, which use heavy oils without danger, is the time gained by not having to light fires or to overcome the heating of the engines. A simple working of the pumps in the morning, and the boat is ready for service at a few minutes' notice during the whole day; there is no abnormal heating caused by the boilers; and the radius of action is increased enormously.

The act of sinking below the surface, which would seem "à priori" more difficult in proportion as the buoyancy on the surface is more considerable, is effected in reality as rapidly in one class as in others.

As regards facility of evolution in a vertical direction during immersion, the submersibles have proved to be more easily handled than the cigar-shaped submarines. In 25 seconds, reckoning from the moment of giving the order, a submersible of 550 tons, already immersed, passes from the immersion of attack (9 meters below the keel) to the immersion of 18 meters, at which point it passes beneath any ship floating on the surface.

When once the 400-ton "Laubeuf" submersibles were adopted, the rôle of passive defence was abandoned, and the Navy now seeks methodically better methods for developing an active and vigorous offensive.

At the present moment this object is more or less attained. The last naval program submitted to the Chamber provides for the construction of 30 new "Laubeuf" submersibles of the improved *Brumaire* type of 400 tons.

The French submarine fleet possesses at present, either in service or in trials, the following units of "Laubeuf" type:

Four *Sirènes* (1901) of 202 tons; *Sirène, Triton, Silure, Espadon.*

Two *Aigrettes* (1904), 172 tons; *Aigrette, Cigogne.*

Two *Circe* (1907), 351 tons; *Circe, Calypso.*

Eighteen *Pluviôse* (1907, 1908, 1909, 1910), 400 tons; armed, in trial, or in construction.

Sixteen *Brumaire* type (1909, 1910, 1911, 1912), 400 tons, in trial or in construction; *Pluviôse* type improved so as to give greater velocity on the surface and in immersion, and with a greater radius of action.

Two improved type *Brumaire* (1911, 1912), 400 tons; in construction. Increase of speed and armament.

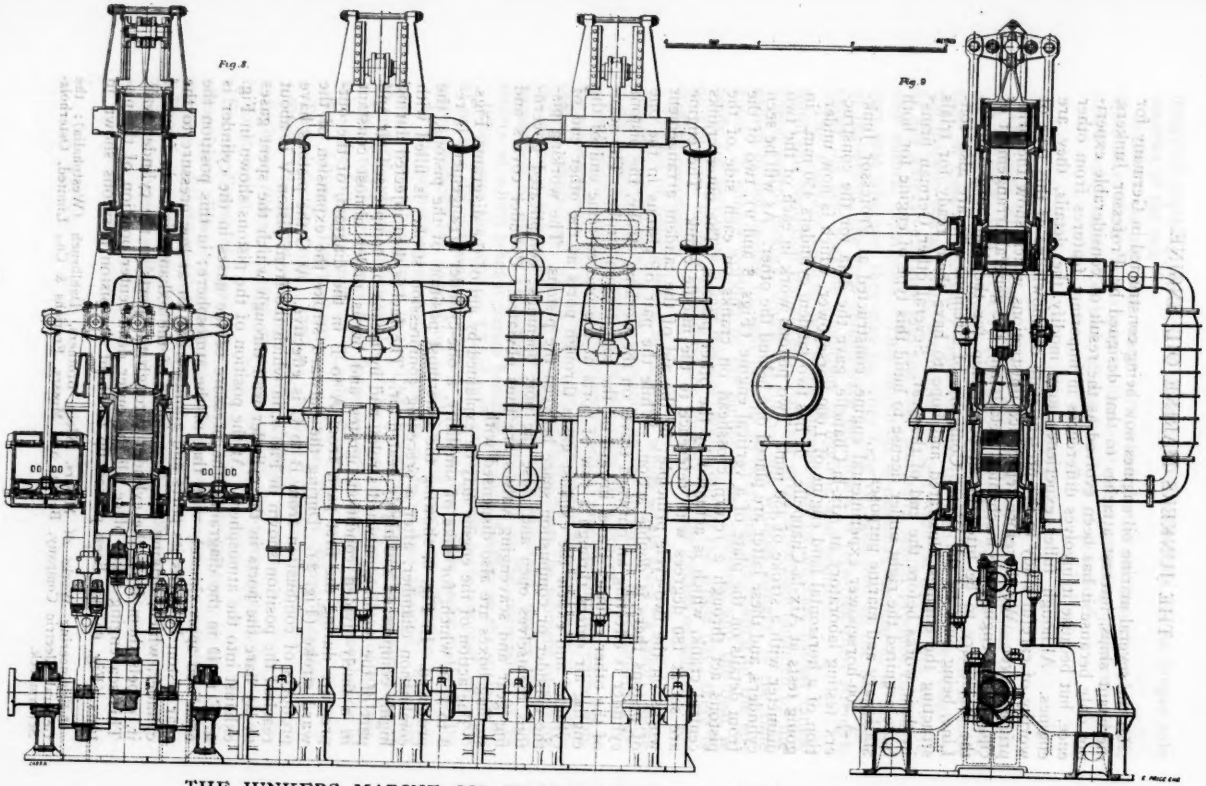
THE JUNKERS MARINE OIL ENGINE.

Of the several marine oil engines now being constructed in Germany for merchant ships, interest attaches to that designed by Professor Junkers, not only because it has been evolved as the result of considerable experience, but because it involves differences in important features from other engines. Although fuller experience may modify the details, they are worthy of careful study in the present state of knowledge of this type of prime mover. We therefore reproduce illustrations which show the principle of the engine, the mechanical details and the general arrangement of the machinery to be fitted in an ocean ship for the Hamburg-America Line, being built by the Weser Company, of Bremen, who are also constructing the engines illustrated and hope to have them ready for trials in their works before the end of the year. Several other German firms* have acquired the right under license to build this type of engine for both stationary and marine purposes.

A 200-horsepower experimental engine, constructed at Professor Junkers' testing laboratory at Aix-la-Chapelle, gave the data for the construction of a horizontal land engine of 1,000-horsepower, which is now undergoing tests at Aix-la-Chappelle. It has two tandem cylinders 450 mm. in diameter with a stroke of 450 mm. Two pistons work in each of the two cylinders, and these latter are placed one behind the other. As will be seen from details on the plate of a vertical engine (Figs. 8 and 9), two of the pistons act through a return crosshead on cranks on each side of the central crank, which is actuated by the other pistons. The outer cranks are set at 180 degrees with reference to the middle crank. The engine works on the two-cycle principle; by the aid of the tandem arrangement of pistons there is double action. While the pair of pistons in the one cylinder is effecting an outward travel on the working stroke, the pistons of the other cylinder travel inward on the compression stroke. The scavenging air enters through a row of ports at the one extreme end of the cylinder and expels the exhaust gases through ports at the other end of the cylinder or combustion space between the pistons. The working pistons themselves open and close the port openings; complicated governing gear and scavenging valves are thus obviated. Cylinder covers and stuffing boxes are also dispensed with.

The action of the engine may be explained by aid of the diagrams, Figs. 2 to 7, in which, for sake of simplicity, a one-cylinder arrangement is reproduced. In Fig. 3, showing the innermost position of the pistons, the combustion chamber, after a preceding compression stroke, is filled with highly compressed and highly heated air. The fuel is then injected during part of the ensuing outward stroke, and by means of compressed air and in a finely-divided condition, ignites and burns under almost constant pressure during the first part, from A to B in the diagram, of the outward stroke (Fig. 2). During the outward stroke the expansion of the products of combustion from B to C is effective. At C the pistons have reached the position shown in Fig. 4, in which the front piston V is about to lay bare the ports in the cylinder wall through which the spent gases exhaust into the atmosphere. At the position of the pistons shown in Fig. 5 (C to D in the diagram), the pressure of the gases in the cylinder is equalized approximately with that of the atmosphere; in this position the rear piston H opens its ports and admits fresh air at low pressure to the cylinder, which, driving the remainder of the exhaust gases in front of it, expels them from the cylinder through the ports in the cylinder wall. This scavenging takes place till the outer dead-center position of the pistons (Fig. 6) has been passed up to about the piston positions shown in

* Engineering Works of Messrs. Klein Brothers, Dahlbruch (Westphalia); the General Electric Company, Berlin; and Messrs. Frerichs & Co., Limited, Osterholz-Scharmbeck.



THE JUNKERS MARINE OIL ENGINES FOR A HAMBURG-AMERICA LINER.

CONSTRUCTED BY THE ACTIEN-GESELLSCHAFT "WESER," BREMEN, GERMANY.

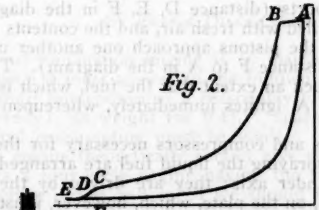


Fig. 2.

Fig. 3.

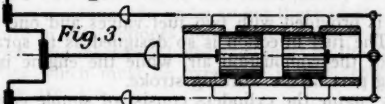


Fig. 4.

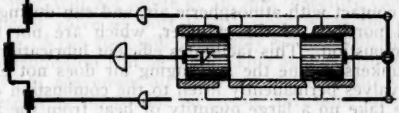


Fig. 5.

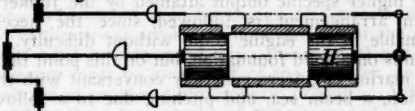


Fig. 6.

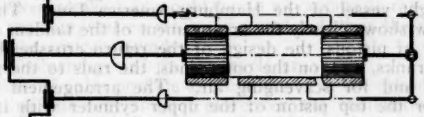


Fig. 7.

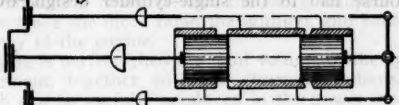


Fig. 7, in which the pistons on the return to the inner dead-center have closed their respective ports (distance D, E, F in the diagram). At the point F, the cylinder is filled with fresh air, and the contents of the cylinder are then compressed as the pistons approach one another up to the inner dead-center in Fig. 3 (distance F to A in the diagram). The compressed air becomes heated to such an extent that the fuel, which is injected at or shortly before the point A, ignites immediately, whereupon the process is repeated.

The scavenging pumps and compressors necessary for the supply of air for scavenging and for spraying the liquid fuel are arranged symmetrically with respect to the cylinder axis; they are driven by the return crosshead, as shown in Fig. 8, on the plate, which, however, illustrates a marine vertical engine.

Each cylinder is provided with two fuel valves and one compressed-air starting valve. The fuel injection is so designed as to spray the oil over a large surface of the combustion air, while the engine is on the dead-center and during part of the working stroke.

In the Junkers engine the cylinders consist of simple castings; the side rods are of forged material. The cylinders rest on simple elastic supports made of I-shaped bars.

The cylinders have no covers; the latter, as is well known, frequently give much trouble in other types of oil motors. The pistons are on one side always in contact with atmospheric air, and run during every outward stroke in cool portions of the cylinder, which are not touched by the products of combustion. This facilitates efficient lubrication of the pistons.

As in the Junkers engine the scavenging air does not enter by the aid of scavenging valves permanently fitted to the combustion chamber, which at every stroke take up a large quantity of heat from the latter and communicate it to the scavenging air rushing through them at a high velocity, the charge, under normal conditions, is much colder than in the ordinary Diesel engine, and this is responsible for the higher mean effective pressure, and the higher specific output attained by the Junkers engine.

The tandem arrangement is followed, since the necessary height is usually obtainable in the engine room without difficulty. Safety against swaying depends on a solid foundation, but on this point there will be some misgiving by marine engineers generally conversant with the influences of engine vibration, a beam sea, and pitching due to a following sea. The cylinders and power-transmitting gear are arranged in the same way as in the horizontal engine, but the general design has been made to suit the vertical arrangement. This design is also followed in its more important features in the engines of twin 800 shaft horsepower under construction for the freight vessel of the Hamburg-America Line. The longitudinal sectional view shows clearly the arrangement of the tandem cylinders, each with its pair of pistons, the design of the return crosshead with its rods to the side cranks, and, on the outer ends, the rods to the pumps for oil-spraying air and for scavenging air. The arrangement of the return crosshead for the top piston of the upper cylinder with its rods to the lower or main crosshead is better seen on the cross-sectional view. The elevation shows the fuel, exhaust and air-supply mains.

For men-of-war the tandem arrangement will have to be dispensed with, and recourse had to the single-cylinder design, owing to limited headroom.

TWO-CYCLE OIL ENGINES.*

By Frank Duncanson.

If the internal-combustion engine is to displace the steam engine for marine propulsion it must answer to the following requirements: It must have a large shaft horsepower at a reasonably low speed (in order to get propeller efficiency), its weight for B.H.P., space taken up, and most important, the fuel consumption must be small.

A two-cycle engine theoretically should give twice as much power at a given speed as a four-cycle engine having the same piston displacement. Hence for a given power and speed we should need only half the piston displacement in a two-cycle engine and consequently should get a lighter and more compact machine. In order to get these ideals realized the mean effective pressure in the cylinders must be the same in a two-cycle as in a four-cycle engine, and mechanical efficiency must be the same in both cases.

The difficulties which must be overcome in the design of a successful two-cycle engine are:

1. Due to imperfect scavenging, the fresh charge is diluted with inert exhaust gases, and hence combustion is imperfect, consequently the mean effective pressure is low.

2. Due to the same cause, weak mixtures, which would otherwise effect an economy in fuel, will not ignite at all, and therefore cannot be used.

3. If the exhaust valve or port is still open when the fresh charge is pumped in, some of the fuel will inevitably escape with the exhaust, and thus the fuel consumption will be high. This applies also to engines which have a "buffer" of scavening air blown in between the exhaust and the mixture, because it is impossible to avoid diffusion, and mixing of the gases due to eddies.

4. The mechanical efficiency of the two-cycle engine will be less than that of the four-cycle engine on account of the auxiliary machinery which must be used in scavenging the cylinder and pumping in the fresh charge.

The conditions to be aimed at in the design of a two-stroke engine are: 1st. At the end of the power stroke we must get rid of the exhaust gases, then the cylinder must be scavenged, *i. e.*, the residual gas now at about atmospheric pressure must be swept out of the cylinder, and the cylinder left full of fresh air at or above atmospheric pressure. While this air is being compressed on the return stroke of the piston and the exhaust ports are closed, a small quantity of fuel must be sprayed into or otherwise intimately mixed with the air to form an explosive mixture which may be ignited towards the end of the compression stroke or at the commencement of the power stroke. It would be an advantage if the fuel were not injected until the compression stroke is completed, thus the danger of pre-ignition would be entirely avoided.

The inherent advantages of the two-cycle are:

1. The absence of valve pockets and big valves in the cylinder head renders the design of this very simple; moreover, the combustion chamber or clearance space is much more compact, thus giving ideal conditions for the ignition of the charge, because, as is well known, the flame originating at the ignition plug takes an appreciable time to travel to the more remote parts of the combustion chamber, and the more compact the combustion chamber can be made the more rapid the ignition, and hence the better the thermal efficiency of the engine.

2. If scavenging is perfect, then the total volume of the cylinder, *i. e.*, the piston displacement, together with the clearance volume, is filled with fresh air which can be usefully employed in the power stroke, whereas in the ordinary four-cycle engine the combustion chamber is always left full

* Abstract of Paper read before the Institute of Marine Engineers.

of exhaust gases, at about atmospheric pressure, which mix up with the fresh charge and dilute it, thus lowering both the capacity and the thermal efficiency of the engine.

3. Much weaker mixtures could be used in well-scavenged two-cycle engines, thus increasing the thermal efficiency to even more than that of a four-cycle engine.

4. A more uniform torque can be obtained with a two-cycle engine because work is done on every downward stroke of the piston. This uniformity of torque, besides enabling a flywheel to be dispensed with, also conduces to better propeller efficiency.

5. The dimensions of the engine would be much less than those of a four-cycle engine of the same power, and this would compensate for the space taken up by the auxiliarics.

By careful design and understanding of the conditions to be aimed at there is no reason why the all-round efficiency of the two-cycle engine should not be as good as or even better than that of the four-cycle engine.

The conclusion to be arrived at is, that to utilize to the fullest extent the advantages of the two-stroke cycle, only independent pumps, which may be designed as such without restriction, should be used.

As regards scavenging, it is of the utmost importance that all the burned gases should be driven out of the cylinder, for on this depends the efficiency, reliability, and capacity of the engine. Assuming that the cylinder volume is to be cleared out with air, the introduction of some air in excess is indispensable, hence there must be available a volume of scavenging air greater than the cylinder volume, because during the scavenging period some air is certain to be lost through the exhaust ports, and unless some excess air is at hand, some of the burned gases are certain to remain. If independent pumps are used, the supply of air may be made as much as is desired, but when the front end of the cylinder or crank case is used to compress the air, an excess is not obtainable.

It is found that as long as the scavenging air is led from the pump directly to the main cylinder, its proper action is seriously hampered. With the pump and main cylinder cranks at 180 degrees apart it is possible to scavenge only at the dead center; where the pump crank leads the main crank, the scavenging air must have comparatively high pressure on account of the short time available for transfer. It seems necessary, therefore, to interpose between pump and power cylinder a receiver of such size that, during the entire scavenging period, the air pressure can be maintained without much drop. This pressure should be as low as is consistent with sweeping out the exhaust gases. An abnormally high scavenging pressure not only increases the lost pump work, but, what is yet worse, interferes with a thoroughly driving out of the burned gases.

By carefully timing the introduction of the scavenging air a great saving in the pumping work might be effected, for the following reason: During the exhaust the velocity of the outflowing gases is very high, in the neighborhood of 2,600 to 3,000 feet per second, and is independent of port area, so that if the exhaust line is made as straight and as long as possible the kinetic energy of the exhaust gas column may be sufficient to overcome all frictional resistances in the line, so that the pressure in the cylinder, owing to the over-expansion of the gas, may drop to less than atmospheric.

When the piston controls the exhaust ports the time taken for exhausting and scavenging should be from 10 to 12 per cent. of the time taken for one stroke, and if the ports are made in a ring round the cylinder a large area is available for the exhaust to get away. It is desirable to have the pressure drop in the air receiver as small as possible in order that the scavenging may continue with undiminished force to the end of the period. To attain this end the receiver should be made as large as possible.

A THIRTY-DAY NON-STOP RUN OF A MARINE OIL ENGINE.

Curle & Co., Limited, are building two sets of eight-cylinder four-cycle Diesel engines of 2,500 combined horsepower for the *Jutlandia*, which they have recently launched, and which is a sister ship of the *Selandia*, which is being engined by Messrs. Burmeiser & Wain, of Copenhagen.

In order to gain some experience of construction and to satisfy themselves on certain points before tackling the actual engines themselves, the makers decided to construct a single-cylinder model, having the cylinder and the main features exactly similar to those proposed for the ship, and to experiment with this. This engine, which has a bore and stroke of 550 mm. and 730 mm.—say, 22 inches and 29¼ inches—respectively, was duly completed some time ago, rigged up on the "bench," and attached to a dynamo, and has demonstrated all the points desired. The culmination of these trials was reached last week, when the engine completed a thirty days' continuous night-and-day non-stop run, of which twenty-eight days were at full power, one day was at three-quarter power and one day at half power, burning Scotch shale fuel oil of a specific gravity of 0.85. During the whole of this period, we are informed, the engine was never stopped or adjusted in any way, and no trouble whatever was experienced, the running being as steady on the last day as it was on the first. Such a run as this should go far to convince the superintending engineer that the Diesel engine has now reached a stage of progress at which it calls for at least his most serious consideration, without asking him to take too much on trust. Certainly there is nothing to prove from the actual running of the engine that the period of thirty days might not have been considerably extended, nor does a careful examination of all the parts on the completion of the run give any indication to the contrary. The engine was stopped at 11 A. M., the 18th January, and dismantling was immediately proceeded with, and by 1 P. M. the valves, piston and connecting rod were lying alongside. Six piston rings are fitted; of these the top ring was found to be gripped in its groove for about one-half of its circumference, and there was some accumulation of dirt round the groove, but curiously enough the whole of the surface of the ring was bright and had the appearance of having been doing its work up to the very last. The second ring was tight, if not actually gripped for about one-third of its circumference, in the same part of the periphery as the top ring, but here again it was bright all over; the rest of the rings were in absolutely perfect condition and were quite free and clean and bright. The outer part of the top of the piston showed a certain amount of accumulation of what was apparently carbonized lubricating oil; the concave part of the top, except in the center, where it was quite bright just where the incoming charge impinged, was coated with a thin film of soot which could be rubbed off with the finger. According to these observations it should apparently have been possible for the piston to have run another thirty days without trouble. The inlet, exhaust and fuel valves were absolutely normal, and these, too, might have run another month.

The makers drew attention in connection with the opening up of the engine after the trial to a curious point with regard to the compressed-air starting valve, a point which we had not heard raised before, though it was anticipated by the builders. They contemplated at the beginning of the trial making it a daily routine to revolve this valve on its seat, but preferred in the interests of experimental data to leave it absolutely alone during the whole of the trial. This valve then had been resting undisturbed on its seat for a period of thirty days in the midst of all the trying conditions of combustion, etc., due to its position in the cylinder head, and it was found that while it could be depressed by the cam gear the spring was not sufficiently strong to return it to its seat, owing to stickiness in the guide.

An examination of the cage showed the presence of a considerable

amount of moisture, rust being evident in places, whether due to the moisture present in the air or to some slight porosity of the metal in the cover it is impossible to say at the moment. In the engines now under construction provision of a special nature is made to meet this requirement, and this provision should be sufficient to obviate such a difficulty in future.

The cams showed absolutely no appreciable sign of wear; indeed, no wear should be expected in ten times the length of run. Judging from the absence of any knocks when the engine was running, beyond those due to the cams striking the rollers, there cannot have been appreciable wear in the pins or bearings. Careful records were kept of all sorts of points throughout the trial, from which we learned that the consumption was 0.45 pound per brake horsepower per hour, the maximum horsepower obtained being 126 at 143 revolutions per minute.

The engine has a trunk piston, though in the eight-cylinder marine type the short piston and crosshead will be adopted; the piston is cooled by oil pumped up the connecting rod, through the gudgeon pin, and overflows into a pipe which leads outside the crank case so as to be under the observation of the engineer; the temperature of this oil was found to keep at about 190 degrees F. The air compressor is driven off the forward end of the crank shaft, and has been deliberately designed with about double the actual capacity required, so as to allow for experiments, etc. This must, of course, tell on the mechanical efficiency of the engine, which should be borne in mind when considering the fuel consumption per brake horsepower, though indeed the latter requires no apology. The compressor is of the two-stage type and delivers air for the fuel injection at a pressure of 64 atmospheres; to effect this the air-inlet valve was practically closed, and when running was remarkably silent. It may be well to describe the reversing arrangements, though they are, of course, practically useless on a single-cylinder engine, and are different from those on the eight-cylinder engine. There are two fuel and two compressed-air cams, each with its own rocker, one pair of each for ahead and one pair for astern, and one inlet and one exhaust-valve cam, each with a rocker. One compressed-air valve rocker and one fuel-valve rocker are mounted excentrically to each other on a common sleeve on the cam shaft in the usual manner, each sleeve being connected to a lever on the starting platform. On receiving the order to go ahead, both these levers being assumed to be in the neutral position, all fuel and compressed-air valves out of action, the ahead lever is raised so as to put the compressed-air valve into operation for ahead, and when the engine is started it is depressed so as to put the fuel valve into operation. This has, however, up to the present, left out of consideration the inlet and exhaust valves, which require in a four-cycle engine to have their relationship to each other altered by roughly a revolution and a-half. This is obtained as in the Blackstone engine, by using the inlet valve as the exhaust valve, and *vice versa*, shunting the fresh air and the spent gases into their proper channels by means of special valves. As the system will not be used in the marine engine we do not feel called upon to comment upon it. The cam shaft is driven by skew gears and a vertical shaft, in the manner now common, the fuel pump and governor being driven off this vertical shaft. The governor, by the way, acts on a special valve which controls the supply of fuel to the suction valves of the pump without interfereing with the actual action of the valves themselves. The whole engine ran remarkably quietly, the compressor and the exhaut-valve cam being responsible for most of what noise there was.

The eight-cylinder engines for the *Jutlandia*, are well on; the cylinders, guides, connecting rods, and covers are all in place; the cam shafts are bedded, and it will probably not be more than six to eight weeks before the engines are running. They will be fitted direct into the ship without any bench tests, a new and rather startling departure, but one which should result in a considerable saving.—"The Engineer."

TABLE SHOWING THE NUMBER AND DISPLACEMENT OF WARSHIPS OF 100 TONS AND UPWARDS LAUNCHED FOR THE VARIOUS NAVIES DURING THE YEARS 1895 TO 1911.

Year.	British.	American (United States).	Austro-Hungarian.	French.	German.	Italian.	Japanese.	Russian.	Other Flags.	Total.	
	tons	tons	tons	tons	tons	tons	tons	tons	tons	No.	tons
1895	139,145	12,034	11,100	42,071	5,490	6,500	2,800	4,114	25,471	82	248,725
1896	117,445	16,302	6,060	57,110	11,100	6,500	24,780	30,281	62,213	92	331,791
1897	66,740	3,000	2,250	15,185	44,214	35,906	18,070	2,200	41,335	90	228,900
1898	140,120	57,900	6,380	25,096	10,648	2,836	45,275	28,650	49,968	91	366,873
1899	121,140	6,400	2,580	52,912	29,240	18,120	61,656	37,230	15,482	91	344,760
1900	35,050	12,330	10,600	40,730	45,330	1,280	26,210	61,840	27,094	99	260,464
1901	209,774	47,903	7,400	40,663	59,400	27,833	1,125	54,680	18,769	123	467,547
1902	92,840	20,449	8,169	44,139	32,324	8,724	8,350	48,485	14,626	102	278,106
1903	147,813	66,140	17,520	30,760	60,590	.....	13,917	38,430	15,930	119	391,100
1904	126,375	170,185	11,480	43,600	44,970	25,932	608	1,750	10,106	102	435,006
1905	96,505	98,200	11,020	28,611	36,487	14,490	50,633	15,721	11,544	118	363,211
1906	85,700	45,443	2,760	15,183	62,678	3,039	41,277	82,204	24,688	142	362,972
1907	133,405	11,590	1,594	33,594	14,800	25,154	57,200	35,317	8,557	142	321,211
1908	49,560	52,850	16,153	21,600	97,660	29,400	2,245	8,800	31,421	127	309,689
1909	98,790	48,639	22,217	95,740	99,116	2,088	375	1,246	36,264	151	404,475
1910	133,525	30,287	14,993	24,063	49,024	19,374	23,100	.....	16,488	122	310,854
1911	221,430	57,526	20,269	53,995	128,340	75,018	37,071	93,260	81,960	169	768,869

—"Engineering."

SHIPS.

UNITED STATES.

BATTLE CRUISERS FOR OUR NAVY.

It is urged by Secretary Meyer that, if we are to prevent a positive decrease in the effective strength of our armored fleet, we must add at least two battleships yearly to our Navy. The older battleships become obsolete through age; the new ships are necessary to maintain our standard.

Also he advocates the construction of a battle cruiser; this vessel to be additional to the two battleships. We consider that a ship of this type should be put in hand at once. The absence of battle cruisers would place us at a very great disadvantage in any operations against a navy that was strong in this particular type. The American Navy should possess ships at least equal to those possessed by foreign powers, class for class. If that principle be admitted, it follows that the magnificent ships of the *Wyoming* and *New York* classes should be represented by one or more 26 to 28-knot, big-gun ships in the battle cruiser class.

It is well known that the construction of armored cruisers of the pre-dreadnought type has fallen off considerably in recent years, but a battle cruiser for Japan was laid down in England last January, and Great Britain has four complete, four launched and two building or provided for, and Germany has two complete, one launched and two building.

Nothing is known of the Japanese vessel, but of the British, six (17,250 to 18,750 tons) are armed with eight 12-inch guns and two (26,350 tons) with eight 13.5-inch. The earliest German ship, *Von der Tann*, displaces 19,100 tons and carries eight 11-inch, but the *Moltke* and *Goeben* are of 23,000 tons and have ten of these weapons.

The principal argument urged against the type is the fact that guns or armor, or both, have to be sacrificed in order to achieve the desired speed, and that high speed itself is unnecessary and confers no advantage upon the ship possessing it except the ability to run away. The objection is undeniably true. A warship is a compromise, and any attribute, such as speed, can be increased only by the reduction of some other, such as armament or armor. This is well illustrated by comparing the British battle cruiser *Lion* with the battleships of the *Orion* class. The latter displace 22,500 tons, have 12-inch belts, are armed with ten 13.5-inch guns, and are designed for 21 knots. The *Lion* is designed for 28 knots; and although she sacrifices two big guns and 3 inches of side armor, her displacement is no less than 26,350 tons, while her total cost, exclusive of guns, is $9,376,220, as compared with $8,849,470 for the *Orion*. These differences are fully accounted for by the fact that the battleship is well served with turbines of 27,000 horsepower, whereas the cruiser requires no less than 70,000 horsepower for her propulsion. The propelling machinery of the *Lion* cost $2,351,165, or 25.1 per cent. of the total cost of the ship (without guns), while that of the *Orion* accounts for only $1,223,265, or 13.8 per cent. of the total. On an average, the machinery of a 21-knot battleship accounts for 16 per cent. of the total cost.

On the other hand, the cost per horsepower in cruisers is considerably lower than in battleships, and is diminishing at a more rapid rate. This

will be seen from the facts that the engines of the battleship *Bellerophon* (1906), of 23,000 horsepower, cost $61.7 per horsepower, and those of the battle cruiser *Invincible* (1906), of 41,000 horsepower, cost $57.6 per horsepower, whereas the engines of the *Orion* (1910), of 27,000 horsepower, cost $45.8, and those of the *Lion* (1910), of 70,000 horsepower, cost $33.6 per horsepower.

The design upon which, for informative purposes, the Bureau of Construction is engaged is said to provide for a battle cruiser with a designed speed of 26 knots and an armament of ten 14-inch guns. Such a ship must be considerably larger than the battleships of the *New York* and *Nevada* classes, which have the same armament, but a speed slower by four knots. The increase would be most pronounced in the length. The British cruiser *Lion* has the same beam as the battleship *Orion*, namely, 88½ feet; but she is 660 feet long, as compared with 545 feet, and this in spite of the fact that she has only four turrets as compared with the battleship's five.

In most dreadnought battleships, American and other, the ratio of length to beam is 6:1, and in most battle cruisers, 7:1. In the case of the *Lion*, however, it is 7½:1. Altogether, it would be a matter of surprise if ten 14-inch guns could be accommodated on a beam of less than 95 feet; and that, according to foreign practice, means a length of between 665 and 712 feet and a displacement of between 31,000 and 33,000 tons, with engines approximating 90,000 horsepower.—"Scientific American."

BATTLESHIPS *NEVADA* AND *OKLAHOMA*.

On January 4, 1912, proposals were opened for the construction of battleships 36 and 37, the *Nevada* and *Oklahoma*. These are to be oil-burning vessels of 27,500 tons displacement and 20.5 knots speed, to be completed within 36 months from date of contract.

The proposals were as follows:

Bidder.	Class.	No. vessel.	Price.	Maximum H.P.	R.P.M.	Type of propelling machinery.
Fore River S. B. Co.	1	1	$5,980,000	24,800	125	Twin-screw reciprocating 4-cyl. triple-ex.
	2	1	5,935,000	26,500	222	Twin-screw Curtis turbine, 132″ H.P. and 138″ L.P., with backing turbine in L.P. casing.
	2	1	5,955,000	26,500	222	Twin-screw Curtis turbine with reciprocating cruising engines.
New York S. B. Co.	1	1	5,965,000	24,100	121.5	Twin-screw reciprocating.
	1	1	5,926,000	24,100	121.5	Do.
Newport News S. B. Co.	1	1	6,450,000	24,800	125	Do.
	1	2	6,350,000	24,800	125	Do.

Class 1, Department design; class 2, contractors' design.

Contract was awarded on January 22 to the Fore River Shipbuilding Company for the *Nevada*, to be propelled by Curtis compound turbines, at $5,895,000, a reduction of $40,000 having been made for the omission of magazine refrigeration. On the same date the New York Shipbuilding Company was awarded the contract for the *Oklahoma*, with reciprocating engines, at $5,926,000.

SUBMARINE TENDER *NIAGARA*.

On March 20, 1912, proposals will be opened for the construction of a submarine tender of about 1,400 tons displacement, with a speed of 14 knots, to be propelled by two-cycle heavy-oil engines.

THE *SACRAMENTO*.

Proposals were opened on February 1, 1912, for the construction of Gunboat No. 19, the *Sacramento*, of about 1,675 tons trial displacement and 16 knots speed, the authorized cost of which, exclusive of armor and armament, is $500,000. The William Cramp & Sons Ship and Engine Building Company were the only bidders. Their proposals were as follows:

Hull and machinery in accordance with Department's design, direct-current electric plant, delivery in 21 months	$716,500
Ditto, except with alternating-current electric plant	723,000
For a vessel of 1,300 tons displacement, and 14 knots speed	485,000

All proposals were rejected.

ARGENTINE REPUBLIC.

THE ARGENTINE BATTLESHIPS *MORENO* AND *RIVADAVIA*.

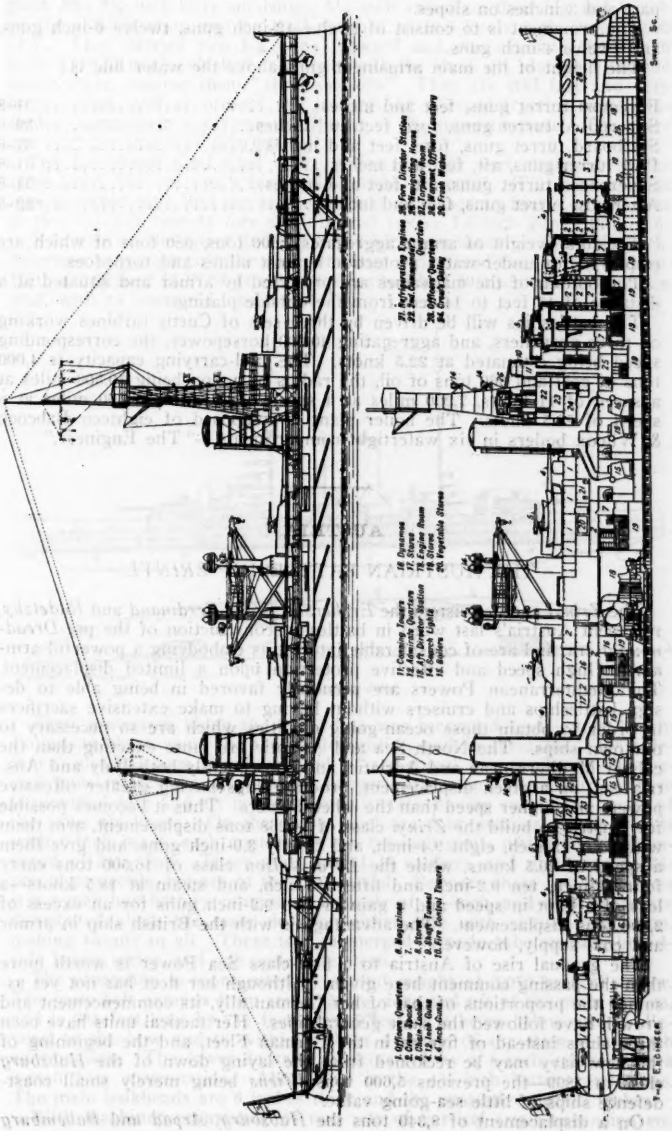
The arrangement of the two Argentine battleships, *Moreno* and *Rivadavia*, is shown in the plate.

The *Rivadavia* was laid down on May 25th, 1910, and was launched on the 26th of August last. The displacement of the vessel is 28,000 tons in speed-trial condition, and the corresponding draught is 27.7 feet. The other dimensions are: Length, 580 feet; breadth, 95 feet 6 inches; depth, 49 feet 7 inches.

The *Moreno* and *Rivadavia* are protected by a water-line belt of armor extending from stem to stern, the thicknesses of which are: 12 inches amidships, for a length of 250 feet and a depth of 4 feet 7 inches above and 3 feet 5 inches below the water line, tapered down to 10 inches thickness at a depth of 5 feet 10 inches below the water line. Fore and aft of the main belt the thickness of the water-line belt is reduced to 10 inches as far as the foremost and aftermost 12-inch gun mounting, and from these points the thickness is 6 inches forward and 4 inches aft.

The citadel armor between the armored deck and the upper deck has a height of 9 feet 6 inches and a thickness of 9 inches below and 8 inches above the upper deck for a length of 400 feet amidships. The thickness of the armor forward and aft of these points is 6 inches and 4 inches respectively.

The casemate armor extends from the foremost 12-inch barbette to the aftermost barbette, and has a height of 8 feet 6 inches and a thickness of 6 inches. The forward conning tower is 12 inches thick, the after tower


1. Officers Quarters
2. Steering Gear
3. Chain Locker
4. 12 Inch Guns
5. 6 Inch Guns
6. Magazines
7. Stores
8. Shaft Tunnel
9. Fire Control Towers
10. Dynamos
11. Stores
12. Ammunition Hoists
13. Stores
14. Officers Quarters
15. Searchlights
16. Boilers
17. Refrigerating Engines
18. Coal Bunkers
19. Stores
20. Vegetable Stores
21. Refrigerating Engines
22. 2nd Cofferdam
23. Coal Bunkers
24. Crew's Galley
25. Officers Quarters
26. Fore Director Station
27. Navigating House
28. Wireless Telegraph
29. Warrant Officers' Lavatory
30. Fresh Water
Swan Sc.
Eng. Engineer

9 inches. The thickness of the protected deck is 2 inches on the horizontal part and 3 inches on slopes.

The armament is to consist of twelve 12-inch guns, twelve 6-inch guns, and twelve 4-inch guns.

The height of the main armament guns above the water line is:

Foremost turret guns, feet and inches	31–8
Superposed turret guns, fore, feet and inches	39–9
Starboard turret guns, fore, feet and inches	31–8
Port turret guns, aft, feet and inches	31–8
Superposed turret guns, aft, feet and inches	31–8
Aftermost turret guns, feet and inches	22–5

The total weight of armor aggregates 7,600 tons, 680 tons of which are employed for under-water protection against mines and torpedoes.

The bottom of the magazines are protected by armor and situated at a distance of 13 feet to 14 feet from the outside plating.

The battleships will be driven by three sets of Curtis turbines working on three propellers, and aggregating 40,000 horsepower, the corresponding speed being estimated at 22.5 knots. The fuel-carrying capacity is 4,000 tons of coal and 660 tons of oil, the radius of action being 10,500 miles at a speed of 11 knots, 7,200 miles at a speed of 15 knots, 3,600 miles at a speed of 22.5 knots. The boiler plant is composed of eighteen Babcock & Wilcox boilers in six watertight compartments.—"The Engineer."

AUSTRIA.

THE AUSTRIAN BATTLESHIP *ZRINYI*.

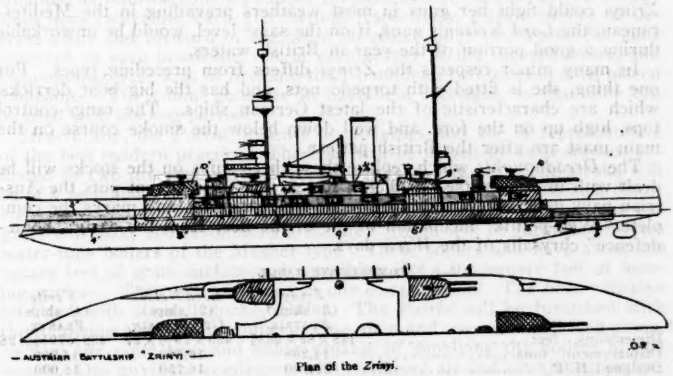
The *Zrinyi* and her sisters the *Erzherzog Franz Ferdinand* and *Radetzky*, represent Austria's last word in battleship construction of the pre-*Dreadnought* era, and are of considerable interest as embodying a powerful armament, high speed and effective protection upon a limited displacement. The Mediterranean Powers are peculiarly favored in being able to design battleships and cruisers without having to make extensive sacrifices in order to obtain those ocean-going qualities which are so necessary to our own ships. The North Sea and Atlantic are more exacting than the calmer Mediterranean and Adriatic, and consequently both Italy and Austria can, on a given displacement, produce ships having greater offensive powers and higher speed than the other Powers. Thus it becomes possible for Austria to build the *Zrinyi* class of 14,268 tons displacement, arm them with four 12-inch, eight 9.4-inch, and twenty 3.9-inch guns, and give them a speed of 20.5 knots, while the *Lord Nelson* class of 16,500 tons carry four 12-inch, ten 9.2-inch and fifteen 3-inch, and steam at 18.5 knots—a loss of a knot in speed and a gain of two 9.2-inch guns for an excess of 2,000 tons displacement. The advantage is with the British ship in armor and coal supply, however.

The gradual rise of Austria to a first-class Sea Power is worth more than the passing comment here given. Although her fleet has not yet assumed the proportions of that of her German ally, its commencement and growth have followed the same general lines. Her tactical units have been three ships instead of five as in the German Fleet, and the beginning of the new navy may be reckoned from the laying down of the *Habsburg* class in 1899—the previous 5,600 tons *Wiens* being merely small coast-defence ships of little sea-going value.

On a displacement of 8,340 tons the *Habsburg*, *Arpad* and *Babemburg*

carried an armament of three 9.4-inch, twelve 6-inch and twenty-six smaller guns, had 8¾-inch belts amidships, 8½-inch armor on their big and 5-inch on the smaller guns, and on trial steamed at 19.6 knots with *circa* 16,000 H.P. They carried two big guns forward and one aft, and the 6-inch guns were in double-decked casements with the small q.f. along the topside above these, making them "three-deckers." They are still fast and very handy ships, and at present are undergoing a reconstruction which includes rearmament and removal of most of the superstructure. The average cost per ship was £650,000.

The *Erzherzog Karl* class which followed them were designed on the same lines, but are much larger and more powerful. The displacement rose to 10,600 tons, and the armament was increased to four 9.4-inch guns in two 9½-inch turrets fore and aft, and twelve 7.6-inch guns, four in 6-inch turrets on the upper and eight in 6-inch casements on the main deck. Fourteen 12-pounders in the superstructure constituted the tertiary battery. With a designed H.P. of 18,000, the ships touched 20.3 to 20.7 f.d. knots on trial, with 19 knots n.d. and 14,000 H.P.



Plan of the *Zrinyi*.

Their protection along the water line was increased to 8½-inch–6-inch. Following these came the trio under notice, and in them the "coast-defence" idea has been abandoned and a "sea-going" type evolved. Twelve-inch guns replace the small-caliber primary armament, and a pair of these is mounted fore and aft in big 10-inch turrets, having substantial bases of the same thickness. On the same level are four smaller 8-inch turrets with 6-inch bases, each containing two 9.4-inch guns, which have a command of 24½ feet. These guns are of 45 calibers, and discharge a 474-pound projectile, but are of the 1907 model. Between these smaller turrets is a 4-inch box battery containing a couple of 3.4-inch, and eight more are distributed along the main deck on either side with similar protection, making twenty in all. There is a submerged torpedo tube on each broadside below water, and one under the stern.

The protection is represented by a total weight of 3,070 tons of armor. The main belt extends from some 15 feet fore and aft of the main turrets, and is 285 feet long by 7½ feet deep, 5 feet of it being below water. Beyond to the bow and to within a short distance of the stem is a 4-inch continuation. The lower-deckside is protected by a 6-inch strake with a 3-inch bow continuation. Above this is the 4¾-inch battery amidships. The main bulkheads are 6 inches thick and the protective deck 2 inches.

With machinery consisting of two sets of vertical triple-expansion en-

gines designed for 20,000 H.P., and actuating two screws, the *Zrinyi* has a legend speed of 20 knots. On trial, over 20,600 H.P. was reached (*Erz. Ferdinand Max*) and 20.76 knots recorded as the highest speed. A normal coal supply of 760 tons can be augmented to 1,350 tons maximum, sufficing for a range of 5,600 miles at 10 knots.

The dimensions are: Length 448 feet (water line), 456 feet (over all), beam 82 feet, and draught (mean) 26½ feet, giving a displacement of 14,268 tons. Herr Siegfried Popper was responsible for the design, and Herrn Gustav Lendecke and Anton Demal for the machinery plans. All the three ships were built and engined by the Stabilimento Tecnico, Trieste, and completed between 1910 and 1911.

From the illustration and plan the general appearance of the ships can be gathered, and it will be noticed how small the actual target is as compared with the *Lord Nelson* type. This is, of course, owing to the fact that the Austrian tertiary battery is along the main deck, whereas in the British ship it is disposed along a flying deck high above the 9.2-inch upper-deck guns. Both systems have their advantages, for while the *Zrinyi* could fight her guns in most weathers prevailing in the Mediterranean, the *Lord Nelson's* guns, if on the same level, would be unworkable during a good portion of the year in British waters.

In many minor respects the *Zrinyi* differs from preceding types. For one thing, she is fitted with torpedo nets, and has the big boat derricks, which are characteristic of the latest German ships. The range-control tops, high up on the fore, and well down below the smoke course on the main mast are after the British pattern.

The *Dreadnoughts* which replace these three ships on the stocks will be dealt with in due course. Suffice it to say that their advent puts the Austrian navy upon a new and more formidable standing, and marks the complete "Welt-politik" incubation of the of the fleet from the early "home-defence" chrysalis of the *Wien* days.

COMPARATIVE TABLE.

	Zrinyi (3 ships) Austria.	*Lord Nelson* (2 ships) Gt. Britain.	*Verite* (4 ships) France.
Dimensions, feet	448×82×26½	435×79½×27	439×79½×28
Displacement, tons	14,268	16,500	14,900
Designed H. P.	20,600	16,750	18,000
Speed, knots	20	18.5	18
Maximum, speed, knot	20.76	18.9	19.46
Armor belt, inches	9–4	12–6–4	11–9–7
side above belt, inches	6	8	10–5½
deck, inches	2	2	2¾
big guns, inch	10	12–8	12½–11
secondary, inch	8	7	5½
Armament—			
Main, inch	four 12	four 12	four 12
Secondary, inch	eight 9.4	ten 9.2	ten 7.6
Tertiary	twenty 3.4" 3 T. Tubes	15 12 pdrs. 16 3 pdrs. 5 T. Tubes	13 9 pdrs. 10 3 pdrs. 4 T. Tubes
Coal supply, tons	750/1350	900/2200 × 400 oil	905/1825
Date of completion	1910–11	1907–8	1907–8

—"The Marine Engineer and Naval Architect."

CUBA.

TWO NEW NAVAL VESSELS UNDER CONSTRUCTION FOR CUBA.

On the 10th of the past October a cruiser and a gunboat for the Island Republic were launched at the shipyard of the William Cramp & Sons Ship and Engine Building Company, Philadelphia.

The gunboat is intended primarily to serve as a training ship for the enlisted men and the cadets of the Navy, and to that end the little vessel will be typically up to date and quite equal to the demands of the practical requirements of Cuba's bluejackets and her embryo officers. The gunboat, which is named *Patria*, has the following principal dimensions:

Length between perpendiculars, feet	185
Beam, molded, feet	34
Depth, feet	22.5
Normal mean draught, feet	12
Displacement, at 12-foot draught, salt water, tons	1,200
Speed on measured mile per hour, miles	16

The armament of the *Patria* will consist of the following guns: Two 6-pounder rapid-fire guns; four 3-pounder rapid-fire guns; four 1-pounder rapid-fire guns; two 7-mm. machine guns.

All of these weapons are of the United States Navy standard, and are mounted as follows: One 6-pounder forward and one 6-pounder aft on the main deck centerline, each gun having, respectively, a train of fire 45 degrees abaft and forward of the beam on either side; two 3-pounder guns mounted on each broadside, one at each corner of the superstructure, and commanding wide arcs of fire, and the 1-pounders, two on each side, placed amidships. The 7-mm. guns are to have field carriages, so that they can be used with landing parties.

Structurally, the *Patria* presents no novelties, but she is representative of the best modern practice. The boat will be driven by twin screws, actuated by two inverted, vertical, triple-expansion engines. The stroke will be of 2 feet, and the cylinders will be of 13-inch, 22-inch and 36-inch diameters, respectively. At full speed, *i. e.*, 16 statute miles an hour, the engines will make 200 revolutions a minute. Steam will be furnished by two water-tube boilers of the Mosher type. These boilers will have about 100 square feet of grate surface and something like 4,400 square feet of heating surface. They will be placed in one compartment. The boat will also be fitted with a small donkey boiler. The *Patria* will be furnished with the following auxiliary machinery: One main and one auxiliary feed pump, each of the same size, and either capable of supplying the boilers at full power. One auxiliary condenser, with combined air and circulating pumps. Two fire and bilge pumps of duplicate dimensions. Two water service and salt water sanitary pumps. One fresh water sanitary pump. One main air pump. One circulating pump. One evaporator and distiller capable of supplying make-up feed and drinking water. These are of the Reilly type. Two forced-draft blowers. One steam ash hoist of the direct-acting steam cylinder type. One ash ejector.

The *Patria* will be provided with accommodations in the deck house amidships for twenty midshipmen and ten cadet engineers. Each stateroom will be arranged to hold from three to four of these young officers.

The cruiser *Cuba* will naturally be of more military value than the *Patria*. Apart from her regular naval service the *Cuba* is designed to be the yacht of the President of the Island Republic. The presidential suite is finished in mahogany, and all of the furniture will be of the same rich wood and Chippendale in pattern.

The *Cuba* has the following principal dimensions and general characteristics:

Length between perpendiculars, feet	260
Beam, molded, feet	39
Depth, feet	26
Normal mean draught, feet	13
Displacement, in salt water, at 13-foot draught, tons	2,055
Speed on measured mile at 13-foot draught, miles	18

She will be armed with the following rapid-fire guns, all of United States Navy standard: Two 4-inch, of 50 caliber; four 6-pounders; four 3-pounders; four 1-pounders; two 7-mm. Colt guns for landing parties.

There will be one 4-inch gun forward and one aft on the main deck centerline, capable of training abaft and forward of the beam, respectively, on either side for 45 degrees. The 6-pounders will be mounted two on each side at the forward and after breaks of the superstructure. The two forward guns will be able to fire directly ahead as well as 60 degrees abaft the beam, while the after pair will be able to fire directly astern and to train 60 degrees forward of the beam. The 3-pounders will be mounted on the main deck amidships, and the 1-pounders will be placed between these weapons.

The 4-inch guns are protected by armored shields, and the armament of the *Cuba* is calculated to make her a very effective craft against filibusters and smugglers; in fact, a thoroughly excellent boat for the police service, for which she is primarily designed.

The *Cuba* is of steel, like the *Patria*. She is provided with a watertight protective deck, which is carried down at the sides below the water line, so as to guard the craft against the admission of large quantities of water in case of injury in the neighborhood of the wind-and-water region. The cruiser is fitted with a double-bottom tank for reserve feed water, and other feed-water tanks will be located in some of the coal bunkers—the compartments being cemented and supplied with all necessary manholes, suction pipes, etc., for the purpose.

Both the *Cuba* and the *Patria* will be lighted electrically and furnished with searchlights. There will be two dynamos and engines of the merchant marine type provided for this purpose. The vessels will be fitted with steering engines of the Williamson Bros. pattern in conjunction with the William Cramp & Sons arrangement for working the cross head on the rudder stock by screw gear.

The *Cuba* is a twin-screw craft, and is to be driven by two inverted, vertical, triple-expansion engines, having cylinders of 16, 26.5 and 44 inches diameter and a stroke of 26 inches. At full speed the engines will make 160 revolutions a minute. These engines are fitted with double-bar Stevenson link reversing gears. All cylinders are fitted with piston valves. One main condenser for the two engines is to be placed in the after part of the engine room; suitable exhaust pipes connecting to both low-pressure valve chests. Two water-tube boilers, Mosher pattern, will furnish the steam. These boilers will be located in a single compartment, and will have 135 square feet of grate surface and 6,000 square feet of heating surface. There will also be a small auxiliary boiler. The boilers will be operated under the closed-stokehold system, with an air pressure not exceeding 2½ inches of water when steaming at full speed. The auxiliary machinery will be a duplicate of that required for the gunboat.

Both the cruiser and the gunboat will have a wireless telegraph system, and will also be fitted with regulation Ardois light signals.—"International Marine Engineering."

DENMARK.

THE DANISH TORPEDO BOAT *SORIDDEREN*.

This vessel is one of the latest additions to the Royal Danish Navy to meet the special conditions of service in the shallow waters of the Great and Little Belt. It therefore possesses many novel and interesting features.

The vessel is 181 feet 9 inches long by 18 feet beam, being larger than the usual torpedo boat; in fact, it is as large as the first destroyers which

were built, but, of course, much smaller than destroyers of the present day. There are ten watertight compartments. One outstanding feature is the raised forecastle forward, and the raised poop aft, thus providing good platforms for the guns and also for working the anchor gear. The armament consists of a 7.5-cm. quick-firing gun forward and another aft; also five 45-cm. torpedo tubes, one of which is built into the structure of the vessel at the bow, the others being on swivel carriages on either broadside. Another point of interest in a vessel of this type is the position of the bridge, which is a long way distant from the bow, and consequently in a much better position for maneuvering the vessel in bad weather, when the spray thrown up by a high-speed vessel makes a position further forward too much exposed. There are two masts, so arranged as to take the aerial wires for the wireless-telegraphy installation, a silent chamber being provided just below the bridge in direct communication with the officer in command.

Steam is supplied by two Yarrow patent water-tube boilers of the latest type, each boiler being placed in a separate compartment. These boilers are fitted for burning coal, and are constructed for a working pressure of 265 pounds per square inch. In these boilers the feed water enters by a special feed pocket in the lower drums, so that it ascends the two outer rows of tubes before mixing with the general circulation of the water.

The main turbines are two in number, of the Brown-Curtis type, built by Messrs. Yarrow & Co., each driving its own shaft, and with an astern turbine incorporated in each main casing, so that each turbine is an independent unit, capable of going either ahead or astern without interfering with the other. For a vessel which has to navigate in narrow channels this system of turbine offers great advantages as regards maneuvering powers. The auxiliary machinery is of the usual type.

The official trials of this vessel were made at Skelmorlie in rough weather, and on the measured mile the mean speed of 28.28 knots was obtained with 5,300 shaft horsepower, the turbines running at 1,050 revolutions per minute. On the three hours' full-speed trial a mean speed of 27.217 knots was obtained, which was well over the contract speed. A consumption trial was also made with the vessel, of which the results obtained were far above the contract requirements, so that the boat has a radius of action, with bunkers full, of 1,400 miles at a 14-knot speed.—"The Engineer."

ENGLAND.

BATTLESHIP TRIALS.

Five battleships, including the *Orion*, have passed successfully through their trials this year, two of them being Portsmouth Dockyard ships, while the others are by the firms named in the table. The conditions attending the trials vary, and it would be a mistake to establish a close comparison between the performances. Moreover, the displacements of the ships differ: the *Orion* and the *Monarch* are of greater displacement than the others and consequently greater power was required to ensure that the same high speed as in the earlier battleships would be realized—namely, between 21 and 22 knots. As typical of speed results, it may be noted that the last of the battleships tried—the *Monarch*—attained a mean of 21.88 knots on the measured-distance runs during the full-power eight hours' trial. The Admiralty are very wisely making practical experiments in all types of ships, in order to ascertain the most efficient propellers; this is most economically done by arranging for differently proportioned propellers in ships of the same class, so that results of running over a lengthened period, under service conditions, can be utilized to collate data. The propeller best suited

STEAM TRIALS OF HIS MAJESTY'S ARMORED AND PROTECTED SHIPS IN 1911.

Name of vessel.	Type of vessel.	Builders of vessel.	Makers of machinery.	Displacement. Tons.	Type of boilers.	Low-power trial.		High-power trial.		Full-power trial.	
						H.P.	Coal per H.P. per hour.	H.P.	Coal per H.P. per hour.	H.P.	Coal or equivalent coal per H.P. per hour.
Neptune.....	Battleship ..	Portsmouth yard	Harland and Wolff.	19,900	Yarrow.....	7,745	3.1	18,373	1.7	27,721	1.4
Hercules	do.....	Palmers.....	Palmers.....	20,000	do.....	12,571	2.4	18,063	2.1	26,599	2.0
Orion	do.....	Portsmouth.....	Parsons	22,500	Babcock & Wilcox.	...	...	18,966	1.8	29,108	1.6
Colossus.....	do	Scotts	Scotts	20,000	do.....	6,454	2.4	19,175	1.7	29,317	1.6
Monarch	do.....	Armstrong Co...	Hawthorn, Leslie & Co.	22,500	Yarrow.....	...	...	19,128	1.8	28,555	1.9
Indefatigable.	Cruiser	Devonport yard.	John Brown & Co.	18,750	Babcock & Wilcox.	10,808	2.1	31,717.9	1.6	47,135	1.5
Falmouth ...	Second-class cruiser.	Beardmore & Co.	Beardmore & Co.	5,250	Yarrow	3,985	1.6	18,374	1.8	26,311.7	1.4
Dartmouth..	do.....	Vickers, Ltd.....	Vickers, Ltd.....	5,250	do.....	4,287	1.7	18,839	1.7	23,467	1.8
Weymouth...	do.....	Armstrong.....	Parsons	5,250	do.....	3,661	1.6	18,076	1.4	23,532	1.4
Blonde.....	Small cruiser	Pembroke yard.	Cammell Laird..	3,350	do.....	4,339	2.0	15.333	1.6	18,770	1.5
Active	do.....	Do	Hawthorn, Leslie & Co.	3,500	do.....	4,044	1.9	15,385	1.4	19,498	1.5

for ships of given size and form is thus determined, and opportunity is taken, in renewing propellers, to fit the ships with the most suitable screws. This is the most reliable method of studying the problem, which can scarcely be solved accurately by the ordinary means of experimenting on models in tanks. As the possible gain in economy is considerable, the ultimate advantage in increased speed for a given power compensates liberally for the expenditure incurred in such trial of different propellers in the ships of a class. As regards the coal consumption of the battleships on the list, it will be seen that on the long trial at what is called high power the average is 1.8 pounds, while on the full-power eight hours' test, when both coal and oil are used, the consumption, stated in terms of coal, averaged 1.7 pounds per shaft horsepower per hour. As every safeguard is adopted to secure service conditions, these results must be reckoned as satisfactory; the fuel required to generate steam for all the auxiliary machinery is included.

The *Indefatigable* is one of the large-powered ships, with a legend speed of 25 knots, which was far exceeded on the trial; in this case the fuel consumption at full speed worked out at only 1½ pounds per shaft horsepower per hour. There are three of the "Town" class of cruisers included in the table. These are ships also of the legend speed of 25 knots, but on trial they each steamed practically 26 knots, and here the consumption, notably at full power, was remarkably low, being 1.4 pounds in the case of two of the vessels. The *Blonde* and the *Active*, built at Pembroke, have already been referred to, and both vessels, like their predecessors, have proved most economical ships, the consumption averaging, at both the high-power and full-power trials, 1.5 pounds. In their case also the speed was practically 26 knots. In the case of all these ships the propelling machinery was of the Parsons turbine type.—"Engineering."

H. M. S. *LION*.

The battle cruiser *Lion*, laid down under the program of 1909–10, has recently completed her steam trials. She is of the following principal dimensions.

Length, 660 feet; beam, 88.5 feet; displacement, 26,300 tons; designed H.P., 70,000; designed speed, 28 knots, armament, eight 13.5-inch, six teen 4-inch, and six smaller guns.

The propelling machinery consists of two sets of turbines, entirely independent of each other, and arranged on each side of the center line of the ship. There are four shafts, each with one propeller. Each set comprises a high-pressure ahead turbine, in which is incorporated a cruising stage at the forward end of the turbine for working at low powers only, the steam being by-passed over this stage at full power or at high fractions thereof; a low-pressure ahead turbine; a high-pressure astern turbine; and a low-pressure astern turbine. The high-pressure ahead and astern turbines are separate, and both are mounted on a wing-shaft, while the low-pressure ahead and astern turbines are within one casing and are on an inner shaft. All of the tubines are of the Parsons reaction type, and the machines in each set work in series. All four shafts are available for ahead and astern working. The engine rooms are suitably divided for protection purposes. Four condensers with steel external plates, with cast-iron covers to protect the tubes from galvanic action, are placed aft, together with the centrifugal pumps and other auxiliaries. The whole of the arrangements, including those of the steam and exhaust piping, feed, drain and oil-lubrication systems, are such as to preserve the independence of the port and starboard sets of machinery, and to allow either set to be worked when all parts of the other are disabled.

The rotor drums, spindles and connecting pieces of the turbine are of forged steel, and the rotor wheels are of forged steel of the arm type. The

turbine and propeller shafting are of forged steel, bored hollow. Thrust and adjusting bearings are fitted to each line of shafting. Complete forced-lubrication arrangements are provided for the turbine and plummer-block bearings. Two main oil-service pumps are fitted in each of the four engine rooms for the forced lubrication of the bearings. A water-service pump is fitted in each of the four engine rooms in connection with the oil coolers.

There are two independent air pumps for each pair of main condensers. The circulating pumps for the main condensers are of the centrifugal type, two for each pair of main condensers. The pumps are driven by independent two-crank engines, fitted with forced lubrication. The fourteen main and auxiliary-feed engines are of the single direct-acting type. One main and one auxiliary-feed engine is placed in each division of the boiler compartments. There are four grease extractors in connection with the delivery pipes of the main air pumps, and grease extractors are also fitted in connection with the auxiliary air-pump discharges. Eleven steam pumping engines are fitted for fire and bilge purposes and for emptying the double bottoms of the compartments in which they are situated. Four of these pumps are fitted in the engine rooms, and the remainder in the boiler compartments. Those fitted in the boiler rooms are also capable of being used in connection with the ash ejectors. In addition to the main condensers there are two auxiliary condensers for taking exhaust steam from all auxiliaries. In connection with these two air and circulating pumps are provided. Six evaporators, worked with steam taken from the auxiliary exhaust system or the auxiliary steam service, are used, and the two distilling condensers are capable of condensing all the steam from four of the evaporators. The remaining evaporators are arranged to deliver the vapor to the auxiliary condensers.

There are forty-two water-tube boilers of the Yarrow type, working at 235 pounds pressure per square inch, and arranged for forced draft with closed stoke-holds. The boilers are fitted in several watertight compartments, with more than usual subdivision. Electrical indicators for regulating boiler firing are fitted, the furnaces being numbered to suit the index of the indicators. Steam-driven air compressors are supplied in each compartment for cleaning the boiler tubes externally by means of an air jet, and for supplying air over the fires. There is a fan for supplying draft to each boiler. They are of the single-breasted type, and are driven by double-acting steam engines fitted with forced lubrication. The boiler room, fire and bilge pumps are adapted for use as ash-ejector pumps.—"Engineering."

NEW TYPE OF SUBMERSIBLE BOAT FOR THE BRITISH NAVY.

The Admiralty have placed an order with the Scotts' Shipbuilding and Engineering Company, Limited, Greenock, for the building of a submersible boat of the Laurenti type, as constructed by the Fiat San Giorgio Company, of Spezia. A feature of the Laurenti design is the construction of an outer hull to give the highest propulsive efficiency and reserve buoyancy on the surface, with the minimum of draught, and an inner hull to minimize the internal cubic capacity while ensuring satisfactory conditions when submerged. The double skin, which is braced with stays to ensure the maximum of structural strength, is confined largely to the central part, and the space between the shells up to the water line on surface displacement is utilized to form water-ballast tanks for submergence. Kingston valves are fitted at the turn of the bilge on each side for the flooding of the compartments, and the structure is made sufficiently strong to enable the water to be pumped out without danger of collapse due to the pressure of the sea water on the outer skin; but compressed air can be, and is normally, used for expelling the water when the boat is to return to the surface. Over the central part of the ship there is a double decking, with lat-

tice bracing, and, valves are fitted on each side above the water line through which water automatically enters and leaves respectively for the submergence or emergence of the vessel, which is effected on an even keel. This double decking extends practically from bow to stern. Vertical bulkheads divide the interior into several compartments. The new British submersible boat will be of the twin-screw type, with twin six-cylinder Fiat engines in one engine room, and electric motors, the latter for propelling the boat when submerged. The torpedo tubes will be forward, under the bow, and the storage tubes above. Italy, Sweden, Denmark and the United States have had submersible boats of this design. The Swedish boat is the more notable, as she made a voyage from Spezia to Stockholm without escort. She is 139.4 feet long, 14 feet beam, and 8.2 feet draught, and the voyage was made in stages respectively to Carthagena, Gibraltar, Lisbon, Oporto, Vigo, Ferrol, Brest, Portsmouth, Ymuiden, Kiel and Stockholm, the longest non-stop run—from Spezia to Carthagena—being 790 sea miles.—"Engineering."

H. M. MOTOR BOAT "NO. 64."

This launch has been delivered to Chatham Dockyard, after a course of tests by Admiralty officials. It was built by Messrs. John I. Thornycroft & Co., Limited, and has been fitted with a Diesel engine by the Diesel Engine Company, of 179 Queen Victoria street. The dimensions are: length, 60 feet; beam, 10 feet; draught, 21 inches. The accommodation comprises a fore-peak with access to deck by watertight circular hatch, a commodious fore-cabin, and aft another cabin, fitted up in the usual way for boats for harbor service generally, *i. e.*, for swinging lamps, folding table, glass racks, etc. The whole of the cabin work is of teak, and the compartments are well ventilated. Lockers are arranged under the seats. Right aft is a large watertight well, protected by a deep coaming and white duck awning overhead. This is provided with four self-emptying lifeboat-pattern valves, and covered by gratings of American elm in sections. The well is provided with seats all round. Immediately under the aft well is a space for stores, to which access is provided by watertight steel doors. In practically all respects as regards fittings, etc., the boat complies with usual Admiralty practice. The engine room is amidships; access to it is obtained through folding door or sliding hatch with ladder. This compartment is lighted with glass windows and brass lamps. The machinery consists of one four-cylinder two-cycle Diesel motor of approximately 100 horsepower, which gives the boat a speed of about 10¼ knots. The scavenging pump is driven off the crank shaft at the forward end, and there are two air compressors, one on each side of the scavenging cylinder. Two long cylinders, fixed one on each side of the engine room, and about 18 inches in diameter, form reservoirs for maneuvering, one being used as a stand-by, while the injection-air reservoirs are of the ordinary type, as fitted for stationary work. Each cylinder has four valves—fuel-inlet, starting-air, and two scavenging-air valves; the fuel, starting, and one scavenging valve on each cylinder are operated by a cam shaft on the port side, and the other scavenging valve by the cam shaft on the starboard side. Reversing in this engine is accomplished by altering the times of operation of the fuel-inlet and starting-valves, this being effected by turning a subsidiary shaft through the required angle, a motion which, by means of levers and eccentrics, alters the position of the cam operating the fuel and starting-valve levers. The subsidiary shaft is actuated by the main starting wheel of the engine, and the arrangement is such that the first movement of the hand wheel causes the engine to run on compressed air, further rotation puts two of the cylinders on air and two on oil, while the final position of the wheel brings all the fuel valves into operation and cuts out all the air-starting valves. Reversing from full speed ahead to full astern occupies only a few seconds.—"Engineer."

FRANCE.

TRIALS OF THE FRENCH BATTLESHIP *VERGNIAUD.*

The French battleship *Vergniaud* forms one of the *Danton* class, and she has recently completed her official trials to the satisfaction, so we understand, of the French Admiralty. There are six vessels in this class, namely, the *Danton, Mirabeau, Diderot, Condorcet, Voltaire* and *Vergniaud.* Their normal designed displacement is 18,400 tons, their length over all 480 feet, their length on the water line 475 feet 9 inches, their beam 84 feet 9 inches, and their maximum draught 27 feet 6 inches. They carry four 12-inch guns and twelve 9.4-inch guns. They are fitted either with Belleville or Niclausse boilers, three having one type and three the other. The *Vergniaud* is the last of her class to be added to the French navy. The *Vergniaud* was built at the Gironde Works, engined by the Mediterranean Works, and fitted with Niclausse boilers. The engines are two sets of Parsons turbines driving four shafts.

We give below the official figures obtained on the trials:

Ten-Hours' Full-Speed Trial.

Date	31st Oct., 1911
Number of revolutions per minute, mean	285.57
Average speed, knots	19.15
Coal consumption per mile run, pounds	2,116
hour, tons	18.09
square meter of grate, kilos	127.80
square foot of grate, pounds	26.17
Steam pressure at boilers, pounds	239
steam chest, pounds	109 to 121
Vacuum at condensers, mm	667

Three Hours' Full-Speed Trial with Forced Draft.

Date	16th Nov., 1911
Number of revolutions per minute, mean	302.96
Average speed, knots	19.675
Coal consumption per mile run, tons	1.17
hours, tons	23.0198
square meter of grate, kilos	161.94
square foot of grate, pounds	33.17
Steam pressure at boilers, pounds	249
turbines, pounds	153.25 to 155.73
Vacuum at condensers, mm	561 to 634

Twenty-four Hours' Consumption Trial.

Date	11th Nov., 1911
Number of revolutions per minute, mean	261.2
Average speed, knots	17.74
Coal consumption per mile, pounds	1,516
hours, tons	12.006
square meter of grate, kilos	84.72
square foot of grate, pounds	17.35
Steam pressure at boilers, pounds	223.2
turbines, pounds	172.0
Vacuum at condensers, mm	602

Consumption Trial with Half Boilers at Work.

Date	7th Nov., 1911
Number of revolutions per minute, mean	182.35
Average speed, knots	13.38
Coal consumption per mile, pounds	959
hours, tons	5.73
square meter of grate, kilos	80.56
square foot of grate, pounds	16.67
Steam pressure at boilers, pounds	216
turbines, pounds	67.8
Vacuum at condensers, mm	615

Slow-speed Consumption Trial.

Date	
No. of boilers at work	7.0
revolutions per minute, mean	119.7
Average speed, knots	8.57
Coal consumption per mile, pounds	710
hours, tons	2.72
square meter of grate, kilos	71.25
square foot of grate, pounds	14.6
Steam pressure at boilers, pounds	228.24
turbines, pounds	22.74
Vacuum at condensers, mm	610

The designed speed of these vessels is 19.25 knots.—"The Engineer."

THE FRENCH DESTROYER *BOUCLIER.*

The French destroyer *Bouclier* was recently handed over to the French Admiralty by the firm of A. Normand & Co., which built her. This vessel is the first among the numerous craft built at the Normand Yard to be fitted with Normand boilers fired only with liquid fuel. The burners employed are of a type patented by the firm. The *Bouclier*, which was engined by the Cie. Electro Mécanique, of Le Bourget, near Paris, has the following particulars:

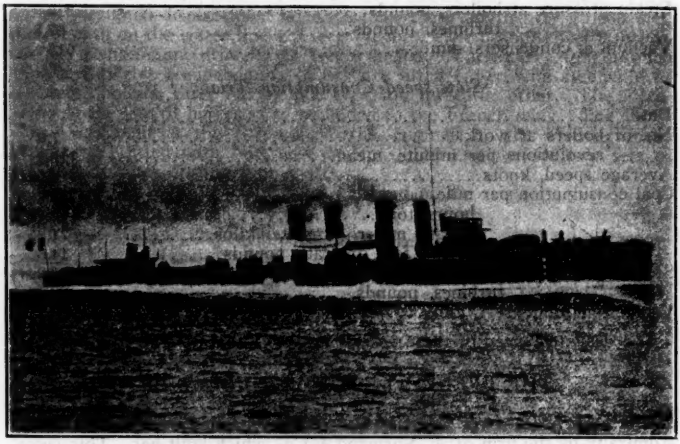
Length over all, feet and inches	233–4
Length between perpendiculars, feet and inches	23–4
Extreme breadth outside plating, feet and inches	24–10
Depth, feet and inches	16–5
Mean draught, feet and inches	12–6
Displacement on trials, tons	660.44

The hull is of high-tensile steel, galvanized in the upper works. It is divided into ten watertight compartments. The hull is clincher riveted, the stem is of forged steel, and the sternpost, etc., of cast steel. The vessel is divided as follows: From forward aft: Collision compartment, crew's quarters with chain locker underneath, ammunition room and storeroom, boiler rooms, engine rooms, officers' quarters, petty officers' quarters, and storeroom. The deck is covered with linoleum and wooden battens.

The *Bouclier* has a high freeboard forward, and owing to this the men are better berthed than in other French destroyers, while the vessel is able to maintain a good speed even in a rough sea. The navigating officer and the man at the wheel are also better protected than is the case in previous boats.

Steam is supplied to the main engines and auxiliaries by four Normand water-tube boilers. The liquid fuel is, before being used, heated in a form of heater which has been newly patented by the Normand firm. The boil-

ers have a heating surface of 5,277 square feet and a combustion chamber of 386 cubic feet capacity. The safety valves are set at 228 pounds per square inch, and at full power an air pressure equal to that of 110 mm.—about 4 inches of water—is permitted in the stokehold. The boilers have been designed each to burn 3,300 kilos. (= 7,273 pounds of fuel) per hour, but as a matter of fact this consumption was not reached during the trials even at full power when the speed attained was actually about 4½ knots in excess of that contracted for.



The main engines consist of Parsons turbines driving three shafts. There are a high-pressure ahead turbine on the center shaft and two low-pressure ahead turbines on the wing shafts. Astern turbines are incorporated in the casings of the two latter turbines. Each shaft carries one propeller, 5 feet 3 inches in diameter and 4 feet 11 inches pitch, designed to work at 1,000 revolutions per minute at full power. The condensers have a cooling surface of 11,682 square feet, and a vacuum of 700 mm. was guaranteed. The four Normand boilers have a heating surface of 21,108 square feet, and the cubic capacity of the furnaces is 1,544 cubic feet. Each boiler has nine burners.

The following are the results of the official trials:

Six Hours' Full-power Trial.

Displacement at start, tons	650.44
Average mean steam pressure, pounds	217
chest pressure, pounds	183
air pressure in stokeholds, mm	109
Pressure in liquid-fuel burners, pounds per square inch	143
Revolutions per minute, mean	1,034.2
Mean speed for six hours, knots	35,334
Contract, speed, knots	31.0
Shaft horsepower	15,000
Vacuum at condenser, mm	700
Consumption of fuel per hour, pounds	21,912
permitted by contract, pounds	29,193
per square foot of heating surface, pounds	1.038
per horsepower hour, pounds	1.46

Eight Hours' Consumption Trial.

Displacement before trials, tons	659.446
Mean draught, feet	12.6
steam pressure at boilers, pounds	214
Revolutions per minute, mean	325.19
Mean speed for eight hours	14.06
as per contract	14.00
Shaft horsepower	1,400
Vacuum at condensers, mm	730.3
Consumption of fuel per hour, pounds	1,915
per horsepower hours, pounds (nearly)	1.37

The shaft horsepower was ascertained by means of a Hopkinson-Thring torsiometer.

The *Bouclier* has up to the time of writing proved herself to be the speediest and most economical vessel of her class, and her performances are certainly excellent. Her armament consists of four 18-inch torpedo tubes. Six torpedoes are carried. There are two 4-inch quick-firing guns, one forward and one aft, and two 2.5-inch quick-firers, two forward and two aft, one on each side. Four hundred and fifty shells are provided for each gun.

The fuel bunker capacity is of such dimensions that the fuel which can be carried permits of a radius of action at 14 knots of 1,950 miles. A view of the vessel under way is given herewith.

ITALY.

ITALY'S FIRST TURBINE-DRIVEN CRUISER, THE *SAN MARCO*.

By Dagnino Attilio.

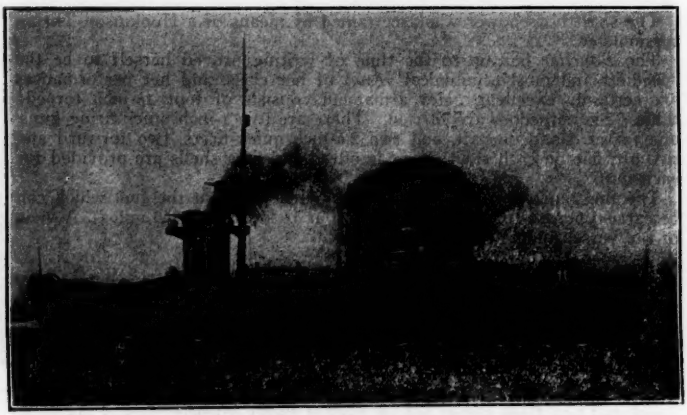
The first Italian turbine cruiser, the *San Marco*, has recently completed a series of very successful steaming trials at Spezia. She was launched at Castellamare, Italy, and engined by Messrs. Ansaldo, Armstrong & Company, of Sampierdarena. She has the following dimensions:

Length over all, feet	461
Length between perpendiculars, feet	427
Extreme beam, feet	69
Depth, feet	39.9
Mean draught, feet	23.6
Maximum draught, feet	24.4
Metacentric height, feet	3.94
Normal displacement, tons	10,110

Her armor consists of a belt, with a maximum thickness amidships of 7.87 inches, decreased to 3.16 inches at the ends, and running from stem to stern with a width of 7 feet 3 inches, of which 4 feet 11 inches is below the waterline. The protective deck has a thickness of 1½ inches. The battery is an extremely powerful one, including four 10-inch guns, 45 calibers long, mounted in pairs in turrets, forward and aft, with an arc of fire of 260 degrees. There are eight 7.5-inch guns, 45 calibers long, in pairs in four turrets at the corners of the superstructure. They have an arc of fire of 160 degrees. The secondary battery includes sixteen 3-inch guns, eight 3-pounders and four Maxims. There are three 18-inch torpedo tubes, all submerged.

The machinery installation of the *San Marco* consists of steam turbines with the usual auxiliary machinery, electric lighting machinery, steam heating and evaporating plants, deck machinery and appurtenances for handling and maneuvering.

The turbines are of the Parsons type, designed for a steam pressure at the turbines of 210 pounds (gage), and will make about 435 revolutions per minute, developing 20,000 shaft-horsepower. There are four shafts, each fitted with one propeller. The installation comprises six ahead turbines, the high and intermediates are used with low-pressure for cruising in order to secure economy, and four backing turbines, two of which are



TURBINE-DRIVEN CRUISER "SAN MARCO."

high-pressure turbines coupled one with each of main high-pressure ahead turbines, and two low-pressure turbines are incorporated into the exhaust ends of each of the low-pressure turbines. The outboard shafts are operated entirely by main high-pressure turbines, the starboard inboard by the low-pressure (starboard) turbine and the intermediate-pressure cruising turbine, and the port inboard by the low-pressure (port) turbine and the high-pressure cruising. In maneuvering the propellers work always in pairs at the same side. All four shafts turn outboard.

The tubine casings are parted horizontally. The lower half carries the box-shaped fore-and-aft bearings, thrust and adjusting block.

The following data of the turbines are interesting:

	Number of expansion.	Mean diameter. Inches.	Length including glands.
Main H.P. port and starboard..	6	49	15 feet 6 inches
L.P. port and starboard.....	6	87	11 " 11⅜ "
H.P. cruising	3	48⅝	11 " 7⅜ "
I.P. cruising	3	55	12 " 3⅜ "
H.P. astern port and starboard..	3	49	7 " 1½ "
L.P. astern port and starboard..	3	64⅞	11 " 11⅜ "

Data of Blades of the Turbines.

	Rows.	Heights. Inches.
Main H.P.—First	13	1
Second	13	1⅜
Third	13	2
Fourth	13	2¾
Fifth	13	4
Sixth	13	5½
Main L.P.—First	5	3 5/16
Second	5	4½
Third	5	6½
Fourth	4	7 5/16
Fifth	4	7 5/16
Sixth	4	7 5/16
H.P. cruising—First	22	5/8
Second	22	13/16
Third	22	1
I.P. cruising—First	20	1 3/16
Second	20	1½
Third	20	2
H.P. astern—First	9	1
Second	9	1½
Third	9	2½
L.P. astern—First	5	3
Second	5	4⅜
Third	5	4⅜

The main condensers are located one in each engine room at the after end of the low-pressure turbines. They are oval and of the surface condenser type. The principal data of each main condenser are:

Number of tubes	8,944
Diameter, outside, inch	5/8
Thickness, inch	3/64
Length of tubes between the plates, feet and inches	7–10¾
Cooling surface, square feet	11,840
Exhaust inlet (rectangular), inches	48½ × 80⅝

There is one auxiliary and one dynamo condenser, located in the forward port engine and dynamo room, respectively. The dimensions of both are:

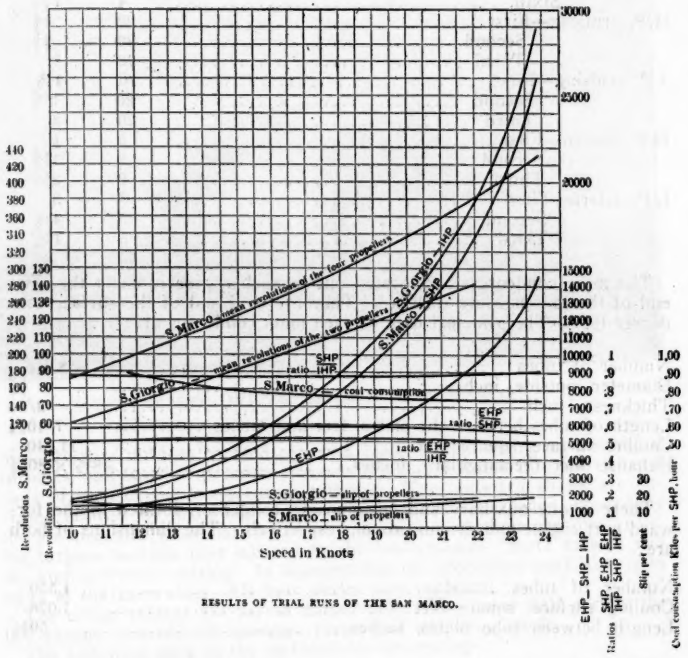
Number of tubes	1,550
Cooling surface, square feet	1,076
Length between tube plates, inches	50⅜

The boiler equipment consists of fourteen units in four firerooms. The total grate surface of all the boilers is 1,409 square feet, and the heating surface 48,201 square feet, the ratio is 34.4 to 1. The working pressure is 250 pounds. Forced-draft system with closed fireroom is used, one blower being provided for each boiler, driven by a vertical, inclosed self-lubricating steam engine.

The official mooring trial, the preliminary progressive trials, the six-hour forced-draft full-power trials, as well as the twenty-four-hour natural-draft trials, were successfully made, and finally a ten-hour contractor's steaming-radius trial was completed.

Experiments were made in using the turbines in three ways. First,

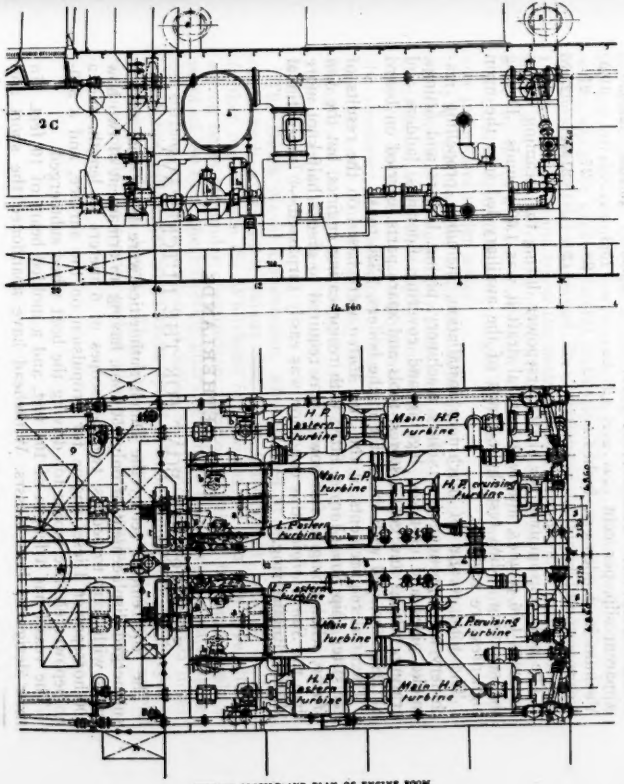
for comparatively low speed with all six ahead turbines in operation; this combination resulted in a smaller quantity of steam being required than with either of the other two combinations. Steam was then admitted initially into the high-pressure cruising turbine, exhausting into the intermediate-pressure cruising turbine, and from the latter through separate pipes to each of the main high-pressure turbines. From these latter steam was exhausted into the low-pressure turbines and finally into the condensers. Second, for speed above 20 knots the five-turbine combination can be used. Steam is admitted initially to the intermediate-pressure cruising turbine, passing thence to the two main high-pressure turbines, and from each of them to the low-pressure turbine. The high-pressure cruising turbine revolves idly in a vacuum. Third, for highest speed only the four main turbines were used, steam being admitted initially to each



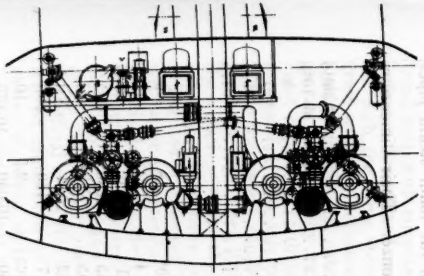
RESULTS OF TRIAL RUNS OF THE SAN MARCO.

main high-pressure turbine, exhausted into the low-pressure and then into the condenser, both cruising turbines revolving idly in a vacuum.

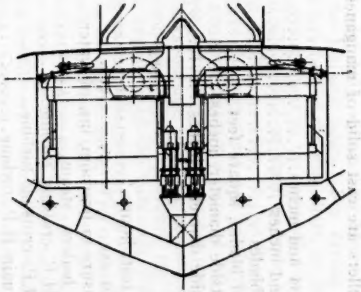
In any of these arrangements the power is governed by throttling, though in the six-turbine combination a by-pass valve is fitted, between the first and second expansion, which may be used, within limits, for this purpose. All maneuvering is done with the four main turbines. All the turbines are drained to the air pump. Glands are bolted to the forward and after casing of each turbine. In order to prevent leakage a steam pressure of about 1 pound above the atmosphere is usually maintained in the gland when cruising by means of a system of piping.



INBOARD PROFILE AND PLAN OF ENGINE ROOM.



SECTION THROUGH ENGINE ROOM.



SECTION THROUGH BOILER ROOM.

a—Main condensers.
b—Main circulating pumps.
c—Main air pumps.
d—Auxiliary condenser.
e—Aux. circulating and air pumps
f—Evaporators.
g—Oil pumps.
h—Oil coolers.
i—Oil cooler pumps.
l—Fire and bilge pumps.
m—Feed pumps.
n—Main feed tanks.
o—Reserve feed tanks.
p—Drain pumps.
q—Engine room aerating fans.
t—Forced-draft fans.
u—Ash hoists.
v—Aux. feed-water filter.

A Cockburn's flexible throttle valve is fitted in the main steam pipes, with an emergency governor gear

The propellers are cast solid of manganese bronze. Their dimensions are:

Diameter, feet and inches	7–06½
Pitch, feet and inches	6–10⅝
Number of blades	3
Total area of blades, square feet	27
Shafting, outside diameter, inches	09¾
Shafting, inside diameter, inches	04⅝

TEST.

Duration of test, hours	10*	18	6
No. boilers in use	4	14	14
Average pressure in fireroom, ins. of water	Nil	Nil	⅝
Pressure of boilers, steam	245	230	245
in H.P. cruising turbine	105	Nil	Nil
I.P. cruising turbine	42	Nil	Nil
main H.P. turbine	17	140	200
L.P. turbines	12 ins. vac.	2.5	10
Displacement of ship	10,050	10,050	10,175
Draught before test, feet		24.2	24.2
after test, feet		23.5	23.95
Average number of revolutions	196.2	374	427.6
Shaft horsepower	2,350	15.700	23,030
Apparent slip, per cent	13.3	.....	19.5
Vacuum in condensers, inches	28	27	27
Average speed, knots	12	21.64	23.75

The coal consumption per shaft horsepower during the steaming radius test was 1.96 pounds, and with natural draft it was 1.60 pounds. During the steaming radius test the exhaust of the auxiliary led into the main high-pressure turbines.

The total weight of machinery installation, including propelling machinery and appendages, auxiliary machinery, piping, boilers and fittings, smoke pipes and up-takes, lagging and covering, floorings, ladders and grating, fittings and gears, stores, tools and spare parts, carried on board is 1,284 tons, and with the water in the boilers, 1,320 tons.

The trial requirements of the *San Marco* were based on the results of her sister ship *San Giorgio*, fitted with reciprocating engines, but the *San Marco*, when going at full power, was required to give a half knot more than the *San Giorgio;* this condition was easily surpassed.—"International Marine Engineering."

THE NETHERLANDS.

SUBMERSIBLE BOAT FOR THE DUTCH NAVY.

The requirements set by the Dutch authorities were for a boat with a submerged displacement of only 150 tons, having an armament of two bow tubes with four torpedoes, two periscopes of 6 meters in length, and a 300-horsepower Diesel engine for propulsion on the surface, and a 300-horsepower electric motor for driving the boat when submerged.

The vessel has a length of 105 feet, and a molded beam of 10 feet. In their design of hull Messrs. Whitehead have abandoned the short and

* Steaming radius. Wetted surface, 1,420 square feet.

broad hull, with circular sections, of the Holland boats, and have adopted instead a longer and more slender hull, with a bow like a torpedo boat, but with the same stern as the Holland boats. For diving, the boat is fitted with two horizontal rudders behind the propellers, as in the Whitehead torpedoes. A patented arrangement in the form of a statical diving gear is arranged, by means of which, and at small expense of power, an exact regulation of the desired depth is possible, so that it can be ensured, for instance, that only the tops of the periscopes will be above the surface.

The pumps are capable of pumping against a head of 200 feet. There are 3-ton drop-keel, lifting eyes, outward connections for pumping air into the vessel or from the air flasks outside, and, for communication when submerged, telephone buoy, submarine signal apparatus, etc.

The 300-horsepower Diesel engine is of the two-cycle Nürnberg type, with starting, reversing and speed regulations arranged to be attained from one single lever. Heavy oil is, of course, used. The accumulator battery is installed in a watertight tank, lined with lead and rubber. Each cell is closed by a cover and can be ventilated, so also can the tank itself. Each cell is independent and can be taken out without disturbing the others. On the official trials the following results were attained, notwithstanding the limited dimensions of the boat: Greatest speed on surface, 11.2 knots; radius of action at 10 knots, 1,000 sea miles; endurance submerged at 7 knots speed, 6 hours 23 minutes; endurance submerged at 8.6 knots speed, 3 hours; endurance submerged at 11 knots, 1 hour. The boat, running at 10 knots on the surface, was tried in all directions in a rough sea, and in a wind of about six force, according to the Beaufort scale, it showed good sea going qualities throughout. It could be navigated from the bridge on the top of the conning tower, and the bow was always well out of the water, while the greatest lateral inclination by rolling was 8 degrees. During the steering trial submerged, the variations in depth could be kept within 1 decimeter above and below the desired depth. Even when running at full speed, and with the rudder moved from the one side hard over to the other side, the depth variation was kept within the limit of a few decimeters. The turning diameters were judged to be about 180 meters on the surface and 160 meters under water.

As a result of the trials the Whitehead Company have obtained nine premiums for having largely surpassed all the given guarantees, the Naval Commission being thoroughly satisfied in every respect. A noticeable feature was the high degree of habitability, which is of great importance for the autonomy of a submarine vessel. Special care has been taken to attain this end. There are three electric cooking ranges, an officers' mess, a living room for the crew, a berth for every man on board, and ample provision of fresh water and victuals, so as to permit a crew of eleven men to live for one week.

The Dutch Naval Commission, who assisted at the preliminary trials, made such a favorable report that their Government decided to place further orders with the Whitehead firm, so that of this new type already eleven boats are under construction. For one of these, having a displacement of 380 tons, the following guarantees have been given: greatest speed on surface, 16 knots; radius of action at 11-knots speed, 2,600 sea miles; endurance submerged at 11-knots speed, 1 hour; endurance submerged at 8-knots speed, 3¼ hours. This boat will also be provided with special arrangements for service in hot climates.—"Engineering."

BOOKS RECEIVED.

INTERNAL-COMBUSTION ENGINE MANUAL. By LIEUTENANT F. W. STERLING, U. S. Navy. Published by the LORD BALTIMORE PRESS, Baltimore, Maryland. 141 pp., illustrated. Price, $1.75 net.

An excellent brief compilation of modern practice, prepared primarily as a text-book for midshipmen at the U. S. Naval Academy.

MARINE STEAM TURBINES. By DR. G. BAUER and O. LASCHE, assisted by E. LUDWIG and H. VOGEL; translation by M. G. S. SWALLOW. Published by the NORMAN W. HENLEY PUBLISHING CO., 132 Nassau Street, New York. 210 pp., 103 illustrations and 18 tables. Price, $3.50.

STRENGTH OF MATERIALS. By JAMES E. BOYD, M. S., Professor of Mechanics, Ohio State University. Published by the MCGRAW-HILL BOOK COMPANY, 239 West Thirty-ninth Street, New York. 290 pp. with illustrations and problems.

ENGINEERING AS A VOCATION. By ERNEST MCCULLOUGH, C. E. Published by DAVID WILLIAMS CO., 239 West Thirty-ninth Street, New York. 201 pp. $1.00.

The author is prominent in his profession, with an experience of more than twenty-five years. The book should prove of value to any one who is investigating the value of an engineering education.

ASSOCIATION NOTES.

KELVIN MEMORIAL.

The Council of the Institution of Civil Engineers have had under consideration the erection in Westminster Abbey of a memorial window to the late Lord Kelvin, as a distinguished engineer long associated with the profession in the advancement of science and its application to enterprises of world-wide importance. They have ascertained that the Dean and Chapter would welcome the addition of such a window to those erected in the north aisles as memorials of eminent engineers, of which that to Sir Benjamin Baker was the latest.

The Council have learned with great satisfaction that many of their professional colleagues in the United States would wish to join in this evidence of respect to the work of an eminent engineer and man of science, and have invited the members of the American Society of Naval Engineers to take part in the provision of the memorial.

The estimated cost of the window is $6,000 and individual subscriptions should not exceed $10.

It is requested that members or associates desiring to participate in this memorial notify the Secretary-Treasurer of the American Society of Naval Engineers.

ANNUAL STATEMENT DECEMBER 31, 1911.

FINANCIAL STATEMENT.

Receipts.

Dues and subscriptions	$5,256.61	
Advertisements	2,531.70	
Interest on bonds	397.50	
Interest on bank account	62.18	
Miscellaneous	1,797.08	
		$10,045.07
Cash balance January 1, 1911		2,740.21
		12,785.28

Expenditures.

Printing and engraving (including wrapping and postage on regular issues of JOURNAL)	$5,349 34	
Manuscript	840.00	
Postage, stationery and expressage	239.38	
Commission on sales, subscriptions and advertisements	121.12	
Salary, secretary-treasurer	900.00	
Salary, clerk	345.00	
Army and Navy Club 5 per cent. bond	1,000.00	
Incidental	1,415.28	
		10,210.12
Cash balance December 31, 1911		2,575.16

Resources.

Cash on hand	$2,575.16
Bonds, 6 Washington Railway and Electric $1,000, 4 per cent. value on market December 31, 1911, 85¼	5,115.00
Bond, 1 Army and Navy Club, 5 per cent	1,000.00
Accounts receivable:	
Dues	300.00
Subscriptions	156.25
Advertisements	607.50
Miscellaneous sales	208.90
Accrued interest on bank account to December 31, 1911	23.52
Furniture: Typewriter, desks, card filing case, etc	51.00
Total	10,037.33

Liabilities, none.

TRADING STATEMENT.

Returns.

Dues and subscriptions	$5,256.61	
Sales	1,221.08	
Advertisements	2,531.70	
		$9,009.39

Costs.

Printing and engraving	$5,349.34	
Drafting	20.35	
Manuscript	840.00	
Postage, stationery and expressage	239.38	
Commission	121 12	
Miscellaneous	214.50	
		$6,784.69
Gross profit		2,224.70

CIRCULATION.

Members.

Number, December 31, 1910		369
Added	46	
Resigned	8	
Died	6	
Dropped	1	
	15	
Net gain		31
Number, December 31, 1911		400

Associates.

Number, December 31, 1910		284
Added	49	
Resigned	7	
Died	1	
Dropped	7	
	15	
Net gain		34
Number, December 31, 1911		318

Subscribers.

Number, December 31, 1910		339
Added	58	
Stopped	18	
Net gain		40
Number, December 31, 1911		379
Total		1,097

Exchanges.

Number, December 31, 1910		75
Added	2	
Stopped	1	
Net gain		1
Number, December 31, 1911		76
Total circulation, December 31, 1911		1,173

Respectfully submitted.

JOHN HALLIGAN, JR.,
Secretary-Treasurer.

Audited and found correct.

R. C. MOODY,
N. H. WRIGHT,
Audit Committee.

The following new members and associates have joined the Society since the publication of the last JOURNAL:

MEMBERS.

Austin, F. L., Second Lieutenant, U. S. R. C. S.
Chambers, W. Irving, Captain, U. S. N.
Clark, C. R., Lieutenant, U. S. N.
Cook, Fayette A., Third Lieutenant of Engineers, U. S. R. C. S.
Corning, M. S., Lieutenant, U. S. N.
Cutter, J R., First Lieutenant, U. S. R. C. S.
Fletcher, Frank F., Rear Admiral, U. S. N.
Griffin, R. M., Midshipman, U. S. N.
Hambsch, Philip F., Ensign, U. S. N.
King, Thomas Starr, 2d, Midshipman, U. S. N.
Oberly, Robert S., Third Lieutenant of Engineers, U. S. R. C. S.
Post, Nathan W., Lieutenant, U. S. N.
Schipfer, Carl A., Ensign, U. S. N.
Shaw, C. H., Lieutenant, U. S. N.
Strother, E. W., Ensign, U. S. N.

ASSOCIATES.

Christy, J. R., Manager, United Engineering Works, Alameda, Cal.
Conti, Angelo, Chief Computing Draftsman, Bureau of Steam Engineering, Navy Department.
Cummings, Henry H., Cummings Machine Co., 110 High Street, Boston, Mass.
DeBaufre, William L., 2035 W. North Avenue, Baltimore, Md.
Dodge, James E., McGill Building, Washington, D. C.
Haynie, W. R., Busch Sulzer Bros. Diesel Engine Co., 30 Church Street, New York City.
Kearfott, W. D., Warren Steam Pump Co., 95 Liberty St., New York City.
Leavitt, William A., Jr., New York Shipbuilding Co., Camden, N. J.
Linch, Charles S., 286 Barrow St., Jersey City, N. J.
Murray, Helge O. H., McGill Building, Washington, D. C.
Powell, H. S., 520 Bond Building, Washington, D. C.
Westinghouse, George, 165 Broadway, New York City.
Wisnom, W. McK., General Engineer Manager, Messrs. Denny & Co. Engine Works, Dumbarton, Scotland.